

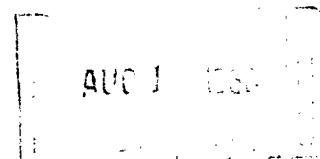
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UNITED STATES
NAVAL POSTGRADUATE SCHOOL



THESIS

THEORETICAL ANALYSIS OF THE HEATED JET
PUMP AND DESIGN OF A TEST FACILITY

by

Wendell Clayton Ridder

and

Carl Richards Summers

May 1966

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THEORETICAL ANALYSIS OF THE

HEATED JET PUMP AND

DESIGN OF A TEST FACILITY

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ABSTRACT

The jet pump is a device that compresses low-pressure fluid by means of a high energy jet. This device is simple, reliable and light weight, and is attractive for various uses including eventual application to boundary layer control for aircraft.

In this study the theoretical flow of a perfect gas through constant pressure and constant area jet pumps is predicted by analyzing the equations of continuity, energy and momentum. Of particular interest is the effect of heating the high energy jet. The parameters describing optimum heated jet pumps are determined. The ideal pumps presented herein represent upper performance limits for actual devices.

Also included is a complete description and design of a facility for testing heated jet pumps. Specific test configurations are analyzed, and performance curves are illustrated. These curves facilitate comparison between theory and experiment.

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TABLE OF SYMBOLS

Symbols

A	cross-sectional area
a^*	speed of sound at Mach Number equal unity
c_p	specific heat at constant pressure
c_v	specific heat at constant volume
C_m	momentum coefficient
C_p	power coefficient
C_{pm}	power-momentum coefficient
C_h	heat coefficient
C_{hm}	heat-momentum coefficient
D	diameter of duct
g	acceleration given a unit mass by unit force
h	static enthalpy
H	total enthalpy
L	length of duct
\dot{m}	mass flow rate
M^*	velocity ratio, V/a^*
p	static pressure
p_t	total pressure
q	heat
R	gas constant
s	entropy
T	absolute temperature
V	velocity

Table of Symbols (Cont'd)

Symbol

x	mass flow ratio, \dot{m}_2/\dot{m}_3
y	area ratio, A_2/A_3
γ	ratio of specific heats, c_p/c_v
Δ	difference or change of
η	efficiency
ρ	density
$\phi(M^*)$	defined velocity function
$\psi(M^*)$	defined velocity function
$()_s$	signifies stagnation state
$()_n$	signifies station location, numbers 1 through 6
$()^*$	signifies the state at which the Mach Number is unity; does not apply to M^*

1. Introduction

A jet pump is a device designed to draw a given mass flow of fluid from a low-pressure source and deliver it to a region of higher pressure by means of a high energy jet of air. Basically, this analysis was undertaken to investigate the theoretical operation of jet pumps, and to recommend a test facility design to provide for the comparison of theoretical and experimental data. The ultimate objective of this investigation is the eventual application of jet pumps to boundary layer control on aircraft.

The jet pump is attractive for an application of this nature due to its reliability, simplicity, and light weight. Also, this device can be readily incorporated into a jet aircraft's power plant cycle. For certain conditions, such as full-throttle operation, it is conceivable that the mechanical power available to the jet pump may be limited. Hence, it is of interest to investigate if the mechanical power required for a given application can be reduced by heating the primary jet air.

The theoretical analysis of ideal jet pump performance contained herein considers the effect of this heating. Constant area and constant pressure mixing processes are investigated. In all cases it is possible to predict "optimum conditions" for specified jet pump configurations. Since all losses except mixing losses are neglected, the results of this study represent a performance limit that actual jet pumps can approach, but never exceed. The final results shown in Appendix C were determined by means of the Control Data Corporation 1604 digital computer. This theoretical analysis is an extension of the work initiated by

Belter in Reference 1.

For any study to be complete it is necessary to substantiate the theoretical results by experimental data. In view of this fact, a design for a heated jet pump test facility is presented. Several test facility configurations along with their material cost estimates are included.

The writers wish to express their appreciation for the assistance and guidance provided by Professor Theodore J. Gawain, of the U. S. Naval Postgraduate School, Monterey, California.

2. Theoretical analysis

Definition of performance coefficients

A jet pump must be able to produce a required discharge momentum. For this output, a certain amount of mechanical power and heat energy must be supplied to the jet air. Reference 1 expresses the power and heat inputs and discharge momentum as dimensionless coefficients. The arbitrary reference dimensions chosen were for ambient air at sonic velocity and the hypothetical area, A_0 , necessary to pass the discharge from the jet pump at sonic velocity.

a. Momentum coefficient

The momentum coefficient is defined as the dimensionless discharge momentum.

$$C_m = \frac{p_s A_s V_s}{p_0^* A_0 a_0^*} \quad (1)$$

b. Power coefficient

The power coefficient is defined as the dimensionless power supplied to the primary air.

$$C_p = \frac{p_s A_s V_s}{p_0^* A_0 a_0^*} \frac{c_p \Delta T_c}{\frac{a_0^{*2}}{2gJ}} \quad (2)$$

where ΔT_c is the temperature change through the compressor.

c. Heat coefficient

The heat coefficient is defined as the dimensionless heat supplied to the primary air.

$$C_q = \frac{p_s A_s V_s}{p_0^* A_0 a_0^*} \frac{c_p \Delta T_p}{\frac{a_0^{*2}}{2gJ}} \quad (3)$$

where ΔT_b is the temperature change through the burner.

Derived coefficients

It is convenient to compare the mechanical power and heat energy to some significant parameter of the jet pump in order to determine the effectiveness of any design. The reference parameter chosen was the discharge momentum of the jet pump. By dividing the power coefficient and the heat coefficient by the momentum coefficient, it is possible to obtain derived power-momentum and heat-momentum coefficients.

$$C_m = \frac{C_P}{C_m} = \frac{\rho_3 A_3 V_3}{\rho_s A_s V_s} \frac{a_s^*}{V_s} \frac{c_p \Delta T_b}{\frac{a_s^* k}{2gJ}} \quad (4)$$

$$C_{qm} = \frac{C_q}{C_m} = \frac{\rho_3 A_3 V_3}{\rho_s A_s V_s} \frac{a_s^*}{V_s} \frac{c_p \Delta T_b}{\frac{a_s^* k}{2gJ}} \quad (5)$$

It is shown in Appendix A that C_{pm} , C_{qm} , and C_{η} can be reduced to the following:

$$C_m = \frac{P_{t2}}{P_{t0}} \left\{ \frac{\gamma+1}{\gamma-1} \left[1 - \frac{P_{t0}}{P_{t2}} \right]^{\frac{\gamma-1}{\gamma}} \right\}^{1/2} \quad (6)$$

$$\frac{C_m}{\left[\left(\frac{P_{t2}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]} = \frac{C_{qm}}{\left[\frac{P_{t2}}{P_{t0}} - \left(\frac{P_{t2}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} \right]} = \frac{\left(\frac{\gamma+1}{\gamma-1} \right) \frac{1}{M_2}}{\sqrt{(x-1) \left(x + \frac{\gamma+1}{\gamma-1} \right)}} \quad (7)$$

where x is the mass flow ratio, \dot{m}_2/\dot{m}_s .

Equation (6) shows that C_m depends only on the discharge total pressure ratio, P_{t2}/P_{t0} . In particular, C_m is independent of the amount of heat supplied. By specifying the desired output pressure ratio, P_{t2}/P_{t0} , the required C_m is determined.

Consider the case where P_{t1}/P_{t0} , T_{t1}/T_{t0} and P_{t2}/P_{t0} are fixed. Equation (7) shows that under these conditions C_{PM} and C_{QM} will be a minimum when the mass flow ratio, χ , is a maximum. It is noted that Equation (7) is invariant for both the constant pressure and constant area cases, although the maximum value of χ may differ in the two cases.

In a sense, C_{PM} is a measure of the compressor power required and C_{QM} is a measure of the heat energy required. Similarly, C_M is a measure of the demanded discharge momentum. Any improvement in design and performance should be reflected by a decrease in power and/or fuel requirements. Therefore, as may be seen from Equation (7), a necessary and sufficient criterion for optimum performance is that the mass flow ratio, χ , be a maximum. Henceforth, an optimum pump is defined as one that has the maximum possible χ for specified values of P_{t1}/P_{t0} , T_{t1}/T_{t0} , and P_{t2}/P_{t0} . It will be shown later that under these conditions an optimum pump corresponds to a specific area ratio, A_2/A_3 . Any other value of area ratio will produce less than the maximum possible mass flow ratio.

Efficiency

Another performance parameter of some interest is the mechanical efficiency which is, of course, based on energy considerations, namely,

$$\eta_M = \frac{\text{Final available energy (Isentropic)}}{\text{Initial available energy (Isentropic)}}$$

$$\eta_M = \frac{\dot{m}_2 (H_2 - h_2)}{\dot{m}_2 \Delta H_c}$$

It is shown in Appendix A that the above definition is reducible to the result:

$$\eta_M = \frac{\left\{ 1 - \left(\frac{P_{t2}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} \right\}}{\left\{ \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}} \left(\chi + \frac{T_{t1}}{T_{t0}} \right) \quad (8)$$

From Equation (8), η_M , like the other performance parameters, will be optimum when χ is a maximum for specified conditions of pressure and temperature ratio.

Constant pressure development

This analysis is an optimization of the performance of the constant pressure heated jet pumps which were described in Reference 1. A typical constant pressure mixing jet pump is shown in Figure 1 along with the associated $h-s$ diagram. The basic development of the constant pressure case is from Reference 2. The many details of the development are contained in Appendix A.

The usual isentropic relationships were used to solve for conditions in the primary (jet) system and in the secondary system up to the mixing tube entrance. It is also assumed that the specific heats remain constant.

The three basic equations that have to be satisfied for constant pressure mixing ($p_2 = p_4$) are continuity, energy, and momentum.

Continuity:

$$\dot{m}_2 + \dot{m}_3 = \dot{m}_4 \quad (9)$$

Energy:

$$\dot{m}_2 H_2 + \dot{m}_3 H_3 = \dot{m}_4 H_4 \quad (10)$$

Momentum:

$$(\dot{m}_2 + \dot{m}_3) V_4 - (\dot{m}_3 V_3 + \dot{m}_2 V_2) = (p_3 - p_2) A_3 \quad (11)$$

It is noted that the resultant pressure forces in Equation (11) are zero when the velocity in the primary nozzle is subsonic.

The equations of continuity, energy, and momentum can be expressed in non-dimensional terms. This development is shown in Appendix A.

Continuity:

$$\frac{A_4}{A_3} = \sqrt{(1+\alpha)\left(\frac{T_{t4}}{T_{t0}} + \alpha\right)} \left\{ \frac{\frac{P_{t4}}{P_{t0}}}{\left(\frac{P_{t4}}{P_{t0}}\right)^{\frac{\gamma}{\gamma-1}} \sqrt{\frac{T_{t4}}{T_{t0}}}} \right\} \left\{ \frac{1 - \frac{\gamma-1}{\gamma+1} M_3^{*2}}{1 - \frac{\gamma-1}{\gamma+1} M_4^{*2}} \right\}^{\frac{1}{\gamma-1}} \frac{M_3^*}{M_4^*} \quad (12)$$

Energy:

$$\frac{T_{t4}}{T_{t0}} + \alpha = (1+\alpha) \frac{T_{t4}}{T_{t0}} \quad (13)$$

Momentum:

$$\sqrt{\frac{T_{t4}}{T_{t0}}} M_3^* + \alpha M_2^* + \sqrt{\frac{T_{t4}}{T_{t0}}} \xi = (1+\alpha) \sqrt{\frac{T_{t4}}{T_{t0}}} M_4^* \quad (14)$$

where,

$$\xi = \frac{1}{\gamma} \left\{ 1 - \left(\frac{P_{t4}}{P_{t1}} \right) \left[\frac{1 - \frac{\gamma-1}{\gamma+1} M_2^{*2}}{\frac{2}{\gamma+1}} \right]^{\frac{\gamma}{\gamma-1}} \right\}$$

By fixing P_{t1}/P_{t0} , T_{t1}/T_{t0} , P_{t5}/P_{t0} , and M_2^* , it is possible to obtain an explicit solution from Equations (12), (13), and (14) since there remains only three unknowns α , T_{t4}/T_{t0} , and A_4/A_3 .

Equations (13) and (14) are solved simultaneously for α and T_{t4}/T_{t0} . By eliminating T_{t4}/T_{t0} between the two equations, the following quadratic equation in α is obtained.

$$(M_4^{*2} - M_2^{*2}) \alpha^2 + \left\{ \left(\frac{T_{t4}}{T_{t0}} + 1 \right) M_4^{*2} - 2 \sqrt{\frac{T_{t4}}{T_{t0}}} M_2^{*2} (M_3^* + \xi) \right\} \alpha + \frac{T_{t4}}{T_{t0}} \{ M_4^{*2} - (M_3^* + \xi)^2 \} = 0 \quad (15)$$

Equation (15) can be solved using the quadratic formula

$$\alpha = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

where

$$a = M_4^{*2} - M_2^{*2}$$

$$b = \left(\frac{T_{t1}}{T_{t0}} + 1\right) M_4^{*2} - 2\sqrt{\frac{T_{t1}}{T_{t0}}} M_2^* (M_3^* + 5)$$

$$c = \frac{T_{t1}}{T_{t0}} \left\{ M_4^{*2} - (M_3^* + 5)^2 \right\}$$

Since the mass flow ratio, α , must be positive and real, one limiting condition is that $b^2 - 4ac \geq 0$. For all cases investigated this condition is satisfied. Other limiting conditions are $M_2^* \leq 1.0$ and $M_3^* \leq 1.0$ since the primary and secondary nozzles are only converging, not converging-diverging. Both roots of Equation (15) were determined. One of these was negative and, therefore, was discarded as meaningless.

From the solution of Equation (15) it was possible to determine T_{t4}/T_{t0} from the energy Equation (13) and A_4/A_3 from the continuity Equation (12). A_1/A_3 can be solved directly from the mass flow ratio.

The computer was used to obtain solutions to Equations (12), (13), and (14). The parameters P_{t1}/P_{t0} , T_{t1}/T_{t0} , R_5/R_{t0} , and M_2^* were varied systematically to obtain families of constant pressure pumps. The optimum pump from each family was determined. Appendix C contains solutions for each optimum pump for the conditions specified.

Constant Area Development

This analysis is an investigation and optimization of the ideal flow of a perfect gas through a constant area jet pump. Figure 2 shows a sketch and a general h-s diagram of a constant area pump. This figure indicates the notation to be used to denote station locations throughout this development.

Much as before, this analysis involves the solution of the continuity, energy, and momentum equations. These equations may be written in absolute form as follows:

Continuity:

$$\dot{m}_2 + \dot{m}_3 = \dot{m}_4 \quad (16)$$

Energy:

$$\dot{m}_2 H_2 + \dot{m}_3 H_3 = \dot{m}_4 H_4 \quad (17)$$

Momentum:

$$\dot{m}_4 V_4 - \dot{m}_2 V_2 - \dot{m}_3 V_3 = p_2 A_2 + p_3 A_3 - p_4 A_4 \quad (18)$$

It is shown in Appendix A that the above equations can be non-dimensionalized to yield the following relationships:

Continuity:

$$y \phi(M_2^*) + \frac{P_{t2}}{P_{t0}} \sqrt{\frac{T_{t0}}{T_{t2}}} \phi(M_3^*) = (1+y) \frac{P_{t4}}{P_{t0}} \sqrt{\frac{T_{t0}}{T_{t4}}} \phi(M_4^*) \quad (19)$$

Energy:

$$y \phi(M_2^*) + \frac{P_{t2}}{P_{t0}} \sqrt{\frac{T_{t0}}{T_{t2}}} \phi(M_3^*) = (1+y) \frac{P_{t4}}{P_{t0}} \sqrt{\frac{T_{t0}}{T_{t4}}} \phi(M_4^*) \quad (20)$$

Momentum:

$$y \psi(M_2^*) + \frac{P_{t2}}{P_{t0}} \psi(M_3^*) = (1+y) \frac{P_{t4}}{P_{t0}} \psi(M_4^*) \quad (21)$$

where:

$$\phi(M^*) = (1 - \frac{\gamma-1}{\gamma+1} M^{*2})^{\frac{\gamma}{\gamma-1}} M^*$$

$$\psi(M^*) = (1 - \frac{\gamma-1}{\gamma+1} M^{*2})^{\frac{\gamma}{\gamma-1}} (1 + M^{*2})$$

and y is the area ratio, A_2/A_3 .

The development of Equations (19), (20) and (21) is from Reference 2. A similar development is included in Reference 1. However, an error

was found in Equation (37) of Reference 1 rendering incorrect solutions for the constant area case presented therein.

It is assumed that the parameters P_{t1}/P_{t0} , T_{t1}/T_{t0} , and P_{t5}/P_{t0} are given. Values of the secondary velocity ratio, M_2^* , are systematically chosen. Once M_2^* is fixed the primary velocity ratio, M_1^* , can be determined as shown in Appendix A. The remainder of the problem resolves into a solution of Equations (19), (20) and (21) for the unknowns M_4^* , γ and T_{t4}/T_{t0} ; and satisfying the boundary condition that the value of P_{t4}/P_{t0} so obtained must equal the value originally specified for P_{t5}/P_{t0} .

Since the resulting relationships are transcendental, it is necessary to solve the equations by iterative procedures. It is thereby possible to achieve solutions for numerous families of constant area heated jet pumps.

The routine employed to solve the equations is to choose incremental values for M_2^* . Choosing M_2^* fixes all the known parameters not previously specified. A trial value of the area ratio, γ , is assumed. The mass flow ratio is then solved by the relationship:

$$\alpha = \dot{m}_2/\dot{m}_1 = \gamma \frac{P_{t2}}{P_{t1}} \sqrt{\frac{T_{t1}}{T_{t0}}} \frac{\phi(M_2^*)}{\phi(M_1^*)} \quad (22)$$

Now it is possible to solve for the discharge total temperature ratio, T_{t4}/T_{t0} , by using the equation:

$$\frac{T_{t4}}{T_{t0}} = \frac{\alpha + T_{t1}/T_{t0}}{\alpha + 1} \quad (23)$$

This procedure determines all the parameters except the exit velocity ratio, M_4^* . The final equation solved is a quadratic in terms of M_4^* .

$$\frac{M_4^*}{1+M_4^{*2}} = \sqrt{\frac{T_{t0}}{T_{t4}}} \frac{4\phi(M_2^*) + (P_{t1}/P_{t0})\sqrt{\frac{T_{t1}}{T_{t0}}}\phi(M_3^*)}{4\psi(M_2^*) + (P_{t1}/P_{t0})\psi(M_3^*)} \quad (24)$$

The above equation gives a supersonic and a subsonic solution for M_4^* . For this particular analysis (i.e., M_2^* and $M_3^* \leq 1$ and constant area mixing), the supersonic solution for M_4^* is meaningless. This means to say that M_4^* cannot be supersonic for a constant area pump when the inlet velocity ratios, M_2^* and M_3^* , are limited to one or less. However, the supersonic solution of M_4^* may be of interest if M_2^* and/or M_3^* are greater than one.

The preceding procedure gives a solution to the equations; however, it is also necessary to satisfy the boundary conditions. The resulting exit total pressure ratio, P_{t4}/P_{t0} , is checked against the desired discharge total pressure ratio, P_{t5}/P_{t0} . If these quantities are not equal, the area ratio is adjusted in order to achieve the necessary conditions at the end of the mixing section. This process is repeated to determine families of constant area jet pumps.

Figure 3 shows the performance curves for two families of constant area pumps. As defined previously, the optimum pumps are those which achieve a maximum mass flow ratio for given values of P_{t1}/P_{t0} , T_{t1}/T_{t0} , and P_{t5}/P_{t0} . It is observed that optimum conditions are reached in two different ways for the constant area case.

Again referring to Figure 3, curve A passes through a true mathematical maximum in the sense that $\frac{dM_4^*}{dx} = 0$. This curve shows that as M_2^* increases the mass flow ratio increases up to a maximum and then decreases. This situation will be referred to as case A hereafter.

Curve B shows the other limiting case of the constant area system.

This curve increases up to a point of maximum mass flow ratio and simply ceases. This phenomena is not clearly understood. Constant area pumps that reach optimum conditions in this manner will be known as case B.

The computer solutions in Appendix C contain a complete set of optimum constant area heated jet pumps for the conditions specified. The non-starred optimum pumps correspond to case A, and the starred pumps are limited in a manner illustrated by case B. These results will be discussed in a following section.

Discussion of Theoretical Results

One significant indication of the performance of a jet pump is the amount of energy that must be supplied to produce the discharge momentum demanded. The coefficients C_{PM} and C_{QM} are simply numbers indicating the energy supplied; and the coefficient C_M is a measure of the discharge momentum required. These coefficients show the effect of changing the parameters P_1/P_0 , T_{s1}/T_0 , and P_{s1}/P_0 .

Figures 4 through 9 are graphs of C_{PM} compared to C_{QM} for families of optimum constant pressure and constant area jet pumps. These graphs are basically maps formed by curves of constant supply pressure and temperature ratios. One of the most significant results illustrated by these maps is the trade-off between the heat energy and compressor power necessary to produce the desired C_M .

For example, consider Figure 4. If the supply pressure ratio is fixed, C_{PM} decreases as the supply temperature ratio increases. This indicates that the compressor power required is reduced by heating the primary air. Conversely, for a constant temperature ratio, C_{QM} is

reduced as P_{t1}/P_{t0} is increased. For the highest temperature ratio and the lowest pressure ratio, the compressor power is minimal. It is also observed that the greatest incremental decrease in C_{PM} occurs between temperature ratios of 1 and 2 along lines of constant pressure.

Now consider the sequence of Figures 4 through 6. It can be seen that as the momentum coefficient demanded is increased, the maps tend to move up and to the right. This, of course, corresponds to higher supply energy requirements. Also, it is noted that the effect of heating is greater as the demand on the pump is increased.

Figures 10 through 15 are graphs of maximum mass flow ratio versus M_2^* . These maps illustrate the performance limitations of the pumps. It must be understood that each point on a map represents a different pump. Therefore, any one curve provides an array of ideal design possibilities. This should not be confused with the performance variation due to changing operating conditions of an individual pump.

Each map shows that as T_{t1}/T_{t0} is increased along lines of constant P_{t1}/P_{t0} , the maximum mass flow ratio increases. This increase is greatest up to a temperature ratio of 2. Beyond this temperature ratio, the effect of heating is less apparent.

Figures 13 through 15 are performance maps for optimum constant area pumps. The lines of constant temperature ratio are continuous; however, there are points of discontinuity in the slope of each line. The dashed curve connecting these points divides each map into two distinct regions. The lower left area of the map corresponds to a flow limitation as shown by curve A of Figure 3. The area to the right of the dashed curve exhibits a flow cut-off similar to that illustrated by case B. These figures also show that the secondary mass flow rate

(\dot{m}_2) decreases as the discharge total pressure ratio increases.

Appendix G contains the solutions of the computer programs for the constant area and constant pressure cases. Each line of data represents the parameters defining one optimum pump. The momentum coefficient is constant for each page of data. Area, velocity, and temperature ratios were included in these results to provide additional criteria for jet pump design. The results show that for a given C_M and T_{t1}/T_{t0} , the discharge total temperature ratio can be decreased by increasing P_{t1}/P_{t0} . This fact could be an important consideration in choosing the material for the mixing tube section.

The important effect of heating the primary air to decrease the compressor power required has been shown. It must be realized that these are idealized cases that represent the maximum performance attainable. Actual jet pumps can approach but never exceed these limits.

Fixed Area-Variable Discharge Pressure

The final theoretical analysis considered is to investigate the operating characteristics of jet pumps with a fixed area ratio (A_1/A_3). By systematically varying the supply pressure and temperature ratios, it is possible to predict the envelope of operation for a given pump configuration. These predicted values are again based on ideal conditions; therefore, they represent the maximum performance attainable.

The fundamental objectives of this analysis are; first, to marry the previously discussed theory to an actual hardware design; second, to determine the design limiting parameters such as size, available compressor power and material limitations; and finally, to predict the performance curves of the jet pumps designed.

The theory of the ideal flow through a fixed area pump is similar to the analysis discussed in the constant area section. However, by fixing the area ratio (A_1/A_2) and allowing P_1/P_2 to vary it is possible to obtain explicit solutions for the equations of continuity, energy and momentum. Appendix C contains an example of the data obtained from the computer program used to predict each pump's performance.

The constant area system is selected primarily due to its relatively simple mixing section design. The supply temperature ratios are limited to three or less, and supply pressures are investigated up to ratios of 2.4. Mixing tube diameters of four, five, and six inches are chosen. This choice is predicated on the nominal pipe sizes available. The primary jet nozzles are designed to give a good spread of area ratios (A_1/A_2). The final result is nine different jet pump configurations. The table below shows the area ratios of the jet pumps investigated.

		NOZZLE EXIT DIAMETER		
		2.00"	2.25"	2.60"
MIXING TUBE DIAMETER	4"	3.000	2.160	1.367
	5"	5.250	3.938	2.698
	6"	8.000	6.111	4.325

Table 1. Area Ratios Selected For Proposed Design

Figures 16 through 24 show the predicted operating maps for the nine

fixed area pumps considered. These curves are prepared primarily for a quick comparison of theoretical and experimental results. Consider the pump with an area ratio of 2.16, shown on Figure 16. Assume that $T_{t1}/T_{t0} = 1.0$, $P_{t1}/P_{t0} = 1.6$ and the back pressure ratio (P_{r5}/P_{t0}) is adjusted to 1.2. Entering the map with these values predicts a mass flow ratio of 1.07. Now if the primary fluid is heated to $T_{t1}/T_{t0} = 3.0$, it can be seen that the mass flow ratio increases to 1.33. This is another example of how heating favorably influences the performance of a jet pump.

It must be clearly understood that these mass flow ratios are simply upper performance limits. Actual jet pumps will not reach these values due to friction and secondary mixing effects.

Theoretical conclusions

For fixed input and output conditions, all performance parameters vary favorably with an increasing mass flow ratio.

For fixed supply and demand conditions, there is one and only one area ratio, A_2/A_3 , that will produce the maximum possible mass flow ratio and hence the best all around performance.

Heating the primary air decreases the compressor power required, i.e., there is a trade-off between heat and power.

The optimum pumps are idealized cases that represent the maximum performance attainable. Actual jet pumps can approach but never exceed these limits.

The ordinary ideas of efficiency are not necessarily decisive in the application of this device to boundary layer control.

3. Jet Pump Test Facility Design

The test facility proposed for studying heated jet pump operation is shown in Figure 25. Basically, it consists of (1) a primary flow system for supplying jet air at a desired pressure and temperature; (2) a secondary air supply system; (3) a set of interchangeable converging jet nozzles; and (4) the constant area mixing tubes.

The test apparatus was designed to be a fixed installation exhausted to the atmosphere. The primary purpose of this facility is to provide fundamental information on jet pump design parameters, and to establish correlation between experimental and theoretical data.

Primary Air Supply System

The primary flow section includes the compressed air supply and fuel supply systems and the combustion chamber assembly. The purpose of the primary system is to provide heated compressed air to the jet pump.

The flow of compressed air to the jet pump is controlled by means of an electrically actuated butterfly valve. It may assume any position between fully closed and fully opened. Thus, this valve controls the pressure of the primary fluid downstream.

After passing through the throttle valve, the compressed air flows into the primary system plenum chamber. The dimensions of this reservoir are arbitrary; however, the chamber must be large enough to serve as a settling tank.

The compressed air leaves the plenum chamber and passes through a square-edged orifice installed in the four inch mild steel exit pipe.

This orifice measures the flow rate of the primary air, \dot{m}_1 . The compressed air is then directed into the burner assembly.

The burner unit is a small canular combustion chamber obtained from an auxiliary gas turbine compressor used for starting jet aircraft engines. The fuel nozzle, igniter plugs and the flame tube are integral parts of the burner unit. The operating range imposed on the jet pump (i.e., $P_{01}/P_0 \leq 2.4$ and $T_{01}/T_0 \leq 3.0$) is well within the capabilities of the combustion chamber.

Fuel is supplied to the burner from a pressurized fuel tank. The fuel pressure is maintained constant by means of a pressurized bottle of inert gas. Fuel flow is manually controlled by a needle valve installed in the fuel supply line. This valve sets the fuel/air ratio and is the primary means of controlling the total supply temperature, T_{01} . The ignition system consists simply of two 12 volt storage batteries connected in series and an ignition control switch.

After the primary air is heated, it passes through a 17 inch section of 5 inch stainless steel pipe. This pipe section is inserted into the secondary air plenum chamber and provides a male fitting for the primary jet nozzle. Total supply temperature and pressure are measured by means of a fixed Kiel-temperature probe. These two values combined with the flow rate are the parameters of prime importance in the primary supply system.

The primary air then passes through a converging nozzle and is ejected into the mixing tube section. The three interchangeable nozzles designed are shown on Figures 26 through 28. These stainless steel nozzles must be machined with close dimensional control and polished to

a mirror finish. They are tapered to ensure uniform flow of primary and secondary air at the mixing tube entrance. A lubricant, such as "silver goop," should be applied to the nozzle threads prior to installation. This material prevents "welding" at elevated temperatures.

Secondary Air System

The secondary air system simply provides a means to measure the secondary air flow rate, \dot{m}_2 . Basically, it consists of an airtight plenum chamber and the associated piping and fittings for a mass flow determination.

Figure 29 shows the proposed secondary air plenum chamber. The reservoir is constructed of 12 gage sheet steel. Standard flanges are used for all piping connected to the chamber. Care must be exercised during fabrication to ensure the airtight integrity of the plenum. Any leaks in the plenum will render the secondary air mass flow measurement inaccurate.

The alignment of the primary air supply duct and the mixing tube section is important. In order to achieve the desired results it is necessary that the jet nozzle and mixing tube centerlines coincide.

Mixing tube

The length of the mixing tube must be such that optimum mixing is accomplished. Three tubes are available with diameters of four, five, and six inches. Each tube consists of four, 24 inch pipe sections which provide for length to diameter (L/D) ratios of 4.3 to 24. L is the length of the mixing tube downstream of the nozzle exit.

The maximum wall temperature attainable is 1100°F; therefore, a heat resistant alloy is advisable. Type 304, schedule 40 stainless

steel which has a melting range of 2550-2650°F and a scaling temperature of 1550°F was selected as a suitable material. With careful attention to the discharge total pressure for fixed supply pressure and temperature ratios, it would be possible to hold the temperature in the mixing tube low enough so that a mild steel or aluminum could be used. This is considered to be an additional experimental burden that can be alleviated by using stainless steel throughout.

One of the simplifying assumptions was to neglect wall friction; therefore, the internal surface of the mixing tube is to be polished to a smooth surface to minimize frictional losses. Consequently, there should be closer agreement between theory and experiment.

By taking temperature and pressure readings along the mixing tube, it is possible to determine where optimum mixing is achieved. Any additional length will only cause a net additional loss in head due to friction. Many of the design questions concerning optimum length of the mixing section can be answered only by experimentation and by investigation of the kinetics of the mixing process in which all effects are considered.

As shown in Figure 27 an oversized flange connects the mixing tube to the secondary air plenum chamber. Attached to this flange is a bell-mouth annulus machined from hot rolled steel. The converging secondary nozzle is formed by this annulus and the exterior of the primary jet nozzle.

The three mixing tubes can be interchanged and may be used with each of the primary nozzles. This flexibility provides for experimental tests of nine different jet pump configurations.

In an actual application, a diffuser would, of course, be used at

the end of the mixing tube to bring the mixed fluids gradually to atmospheric pressure. For the purposes of the present research, separate diffusers were not designed due to the many varying operating conditions and physical configurations.

The mixing tube is supported by three stanchions similar to the one illustrated in Figure 30. Each stanchion has two turnbuckles which allow alignment in the vertical and horizontal planes.

The handling cart shown in Figure 31 will be used whenever it is necessary to change the mixing tubes. Two carts of this design would be desirable to provide the versatility prescribed for the test facility. The carts are made almost exclusively of aluminum channel and angle stock. The maximum design load is 1000 lbs.

Instrumentation

A typical 24 inch mixing section with the associated instrumentation is shown in Figure 32. Each section contains six static pressure taps. These taps are located at four inch intervals along the entire mixing tube. Since the assumption is made that a constant static pressure exists across each cross section, the static pressure is measured only at the mixing tube walls.

Total pressure and total temperature readings are to be taken with a Kiel-temperature probe. These readings can be taken at any of eight different stations along the mixing tube. At each station there are two probe fittings which permit transverse surveys of temperature and pressure along either of two diameters.

The mounting plates shown at each station support the receiver containing the Kiel-temperature probe. The receiver is mechanically

controlled remotely by a transmitter. This transmitter controls the transverse and angular movement of the probe.

When the probe is inserted in one of the probe fittings, the remaining holes are plugged. These plugs are machined to be flush with the inside walls of the mixing tube.

A butterfly valve is installed in a pipe section after the mixing tube to regulate the discharge total pressure. A second Kiel-temperature probe is available to be installed at the end of the mixing tube to measure this pressure.

This test facility provides the means to determine the primary and secondary mass flow rates. In addition, velocity and temperature profiles may be determined along the length of the mixing tube, thereby enabling determination of an optimum mixing length.

Mass flow measurement

It has been shown that one of the most important performance parameters is the mass flow ratio, \dot{m}_2/\dot{m}_3 . Figure 33 shows the installation of two square-edged orifices. The orifice plates for the primary and ambient air supply have holes of 2.5 and 3.75 inch diameter respectively. They are installed in slip-on, butt welded flanges which have pressure taps meeting ASME specifications. The two flange taps are connected to a water filled manometer board where the pressure difference, Δh_w , is measured. The gage pressure P_1 , and temperature t_1 , must be measured ahead of each orifice.

ASME requirements prescribe a minimum straight pipe length ahead of and behind an orifice for a flow rate measurement. The straight pipe length shown in Figure 25 exceeds the minimum requirements; therefore, it is not necessary to install flow straighteners.

Reference 3 contains all necessary equations for determination of the flow rates.

Cost estimate

The cost estimate is based on the following assumptions:

- a. No old material will be used except the burner and fuel supply system, pressure and temperature gages, and manometer boards.
- b. Heat resistant alloy is necessary in the mixing tube.
- c. All machining and fabrication can be accomplished by shop personnel.
- d. Three nozzles and three mixing tubes are necessary to check a range of configurations.

Two cost estimates are provided:

- | | |
|--|-----------|
| a. Total material cost with remote electrical control: | \$5555.42 |
| b. Total material cost with remote mechanical control: | \$4787.42 |

The total man-hour labor estimate is 362 hours.

The material cost could be reduced considerably if only one nozzle and one mixing tube were used. This is shown in the following estimate:

- | | |
|--|-----------|
| a. Total material cost with remote electrical control: | \$3734.08 |
| b. Total material cost with remote mechanical control: | \$2969.08 |

A minimum cost estimate is provided based on the following assumptions:

- a. One nozzle and one mixing tube will be used.
- b. The mixing tube consists of two, 24 inch mixing sections

instead of four.

e. Handling carts are not required.

d. The supply temperature ratio is reduced to $T_{t1}/T_{t0} = 2.0$.

Therefore, stainless steel could be replaced by mild steel.

e. Manual controls are used instead of electrical.

f. As much old material as possible will be used such as plastic tubing, wall taps, copper tubing, connectors, pressure plugs, burner and fuel supply system pressure and temperature gages, and manometer boards.

g. Four inch pipe will be used in the mixing tube.

The total minimum material cost is \$1601.50.

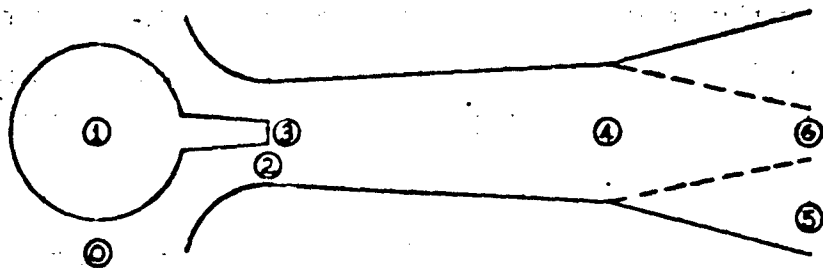
An itemized cost estimate is contained in Appendix B.

4. Recommendations for further study

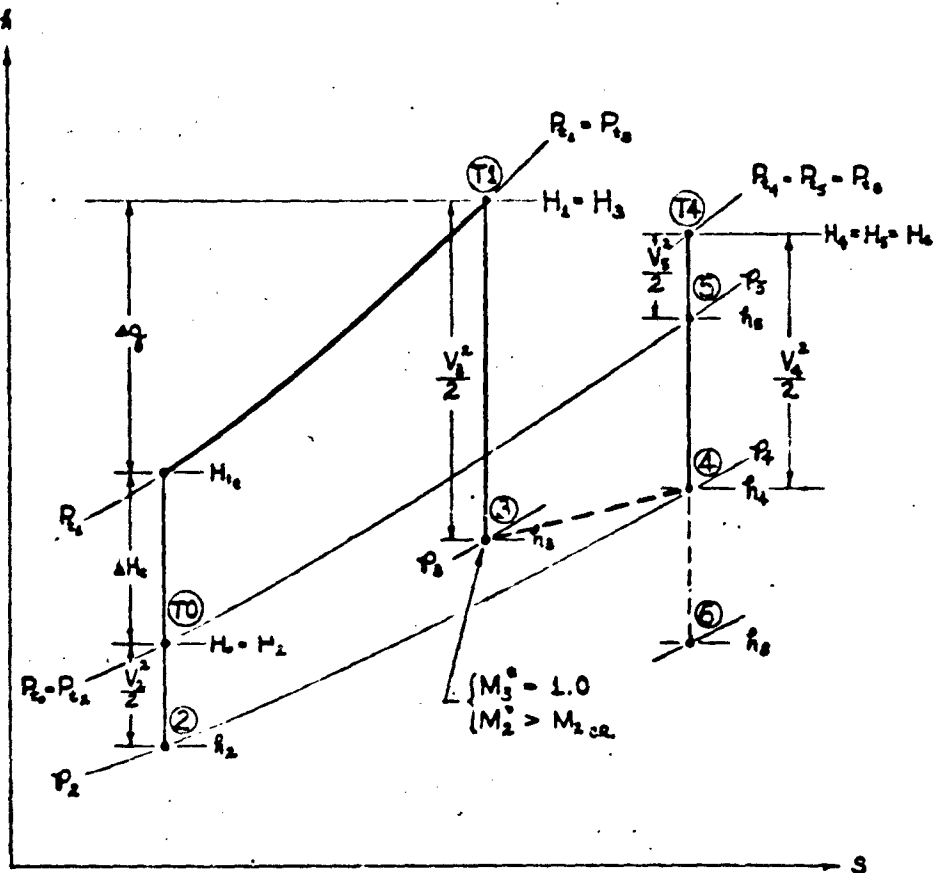
At this stage of the investigation, the most essential requirement is to build the test facility described in this report. If this is not feasible at the present time for economic or other reasons, a further theoretical analysis would be useful to study the effects of wall friction and incomplete mixing. These analytical studies should also include the effects of variable specific heats.

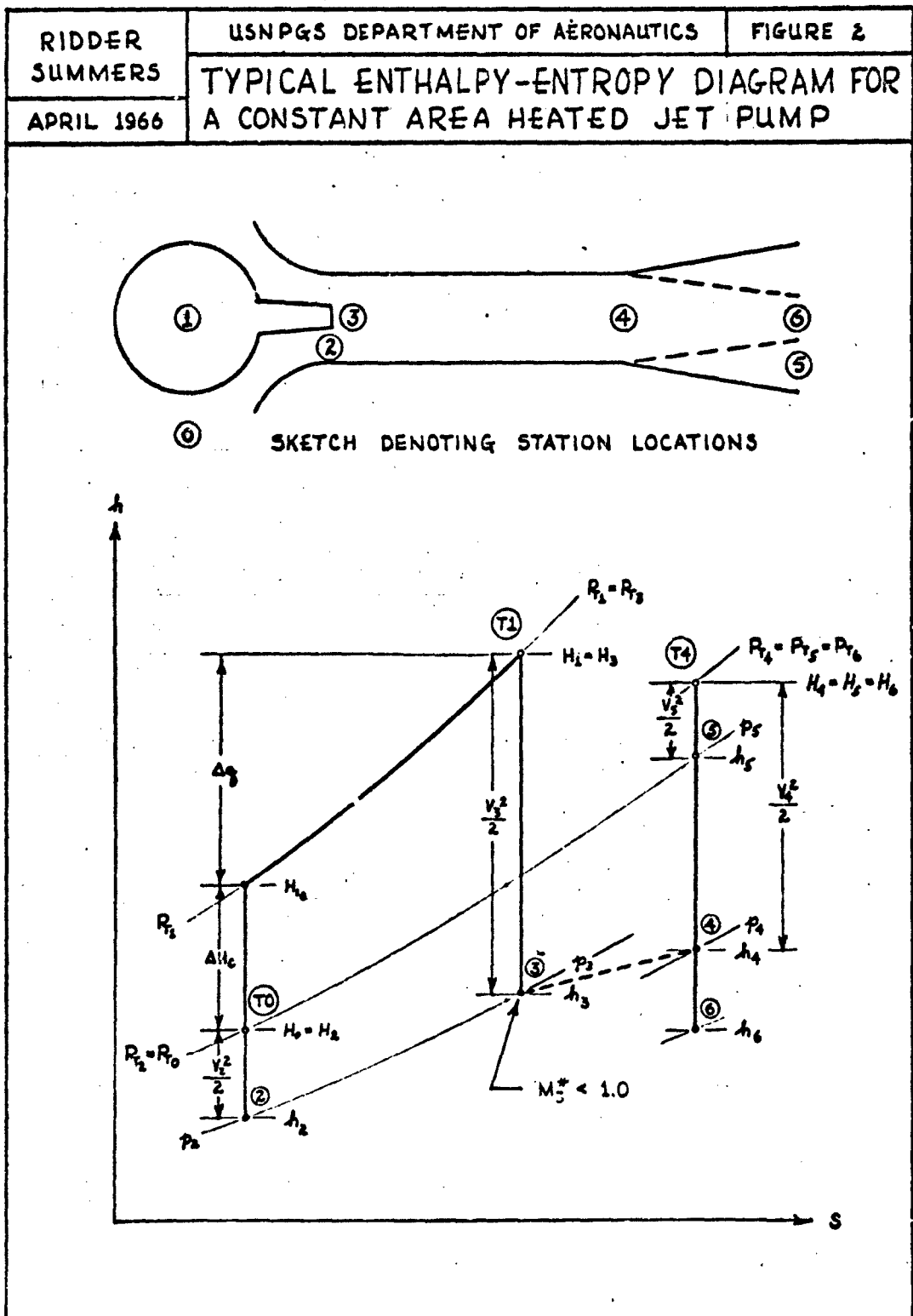
A further recommendation is to proceed with the preliminary design of an experimental boundary layer control system utilizing such a heated jet pump.

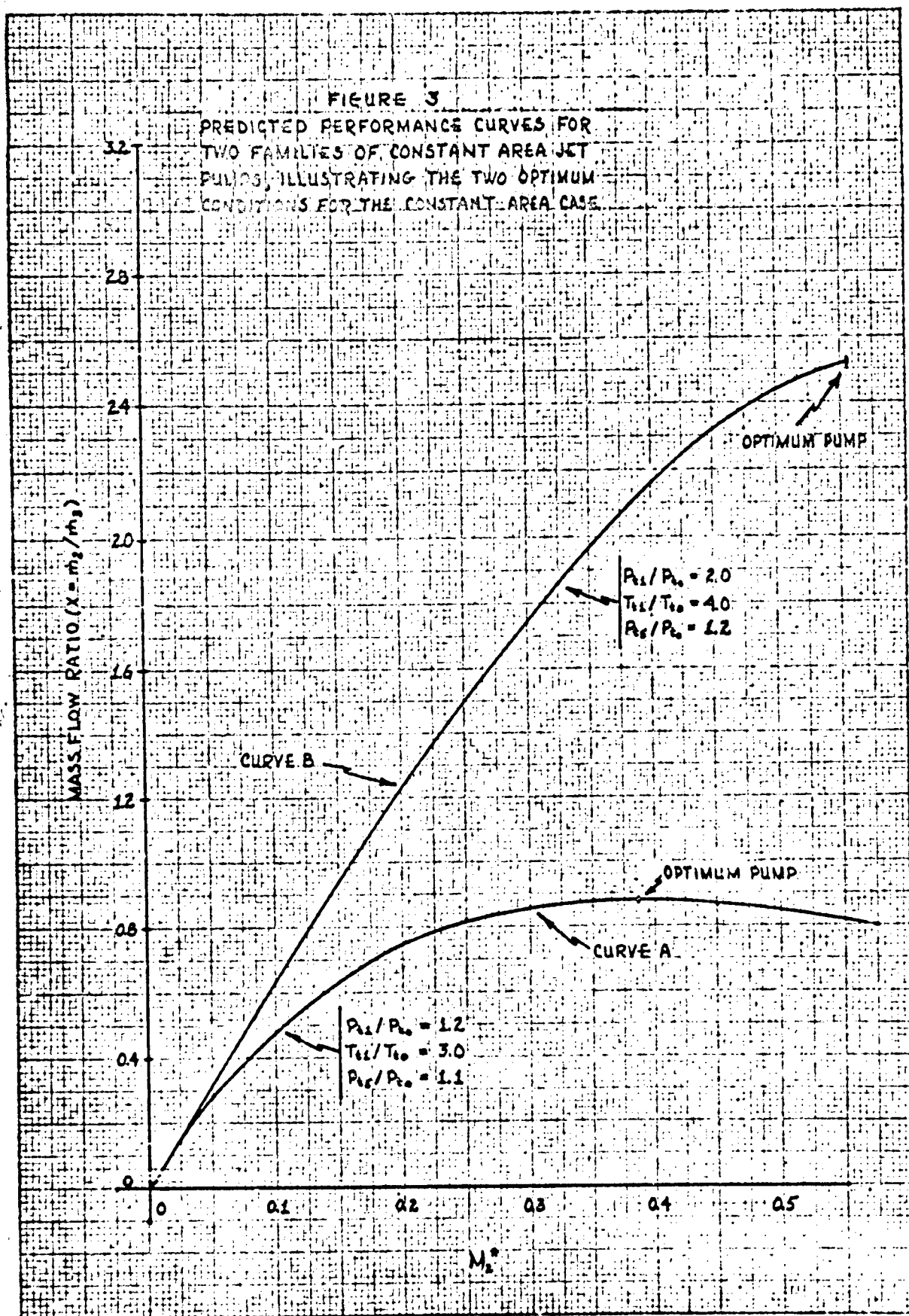
It also would be of interest to investigate the performance of a heated jet pump with a converging-diverging primary nozzle.

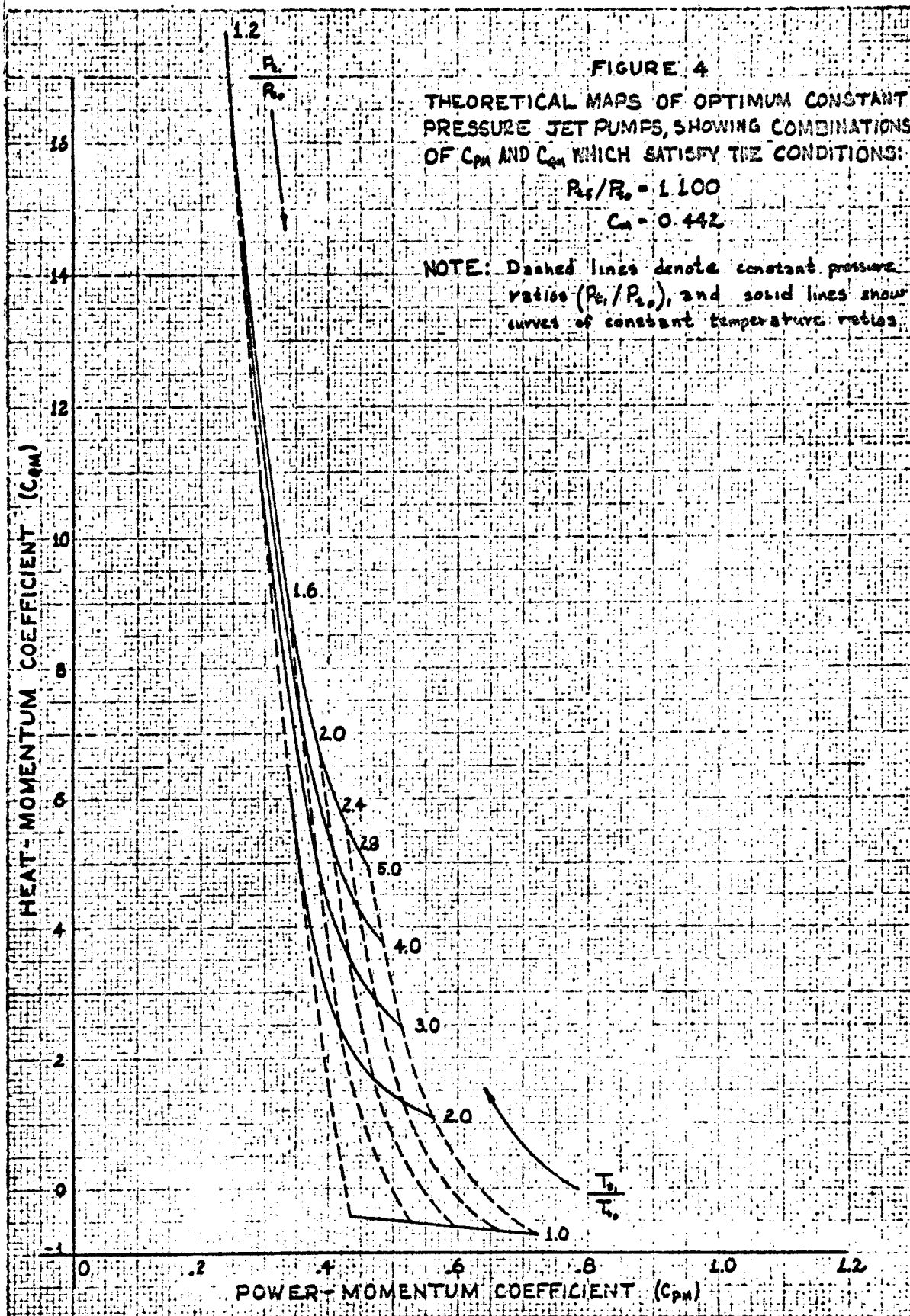


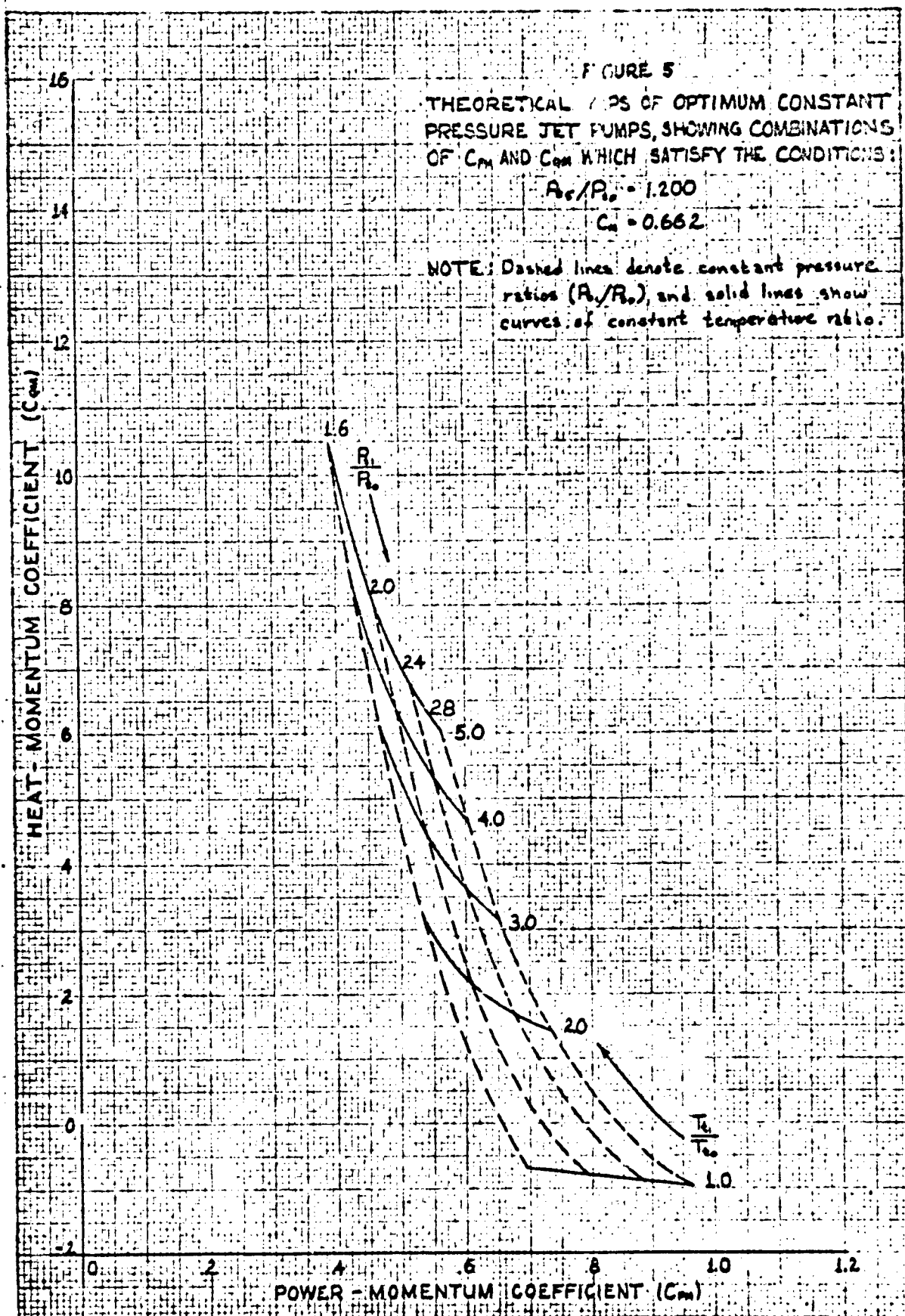
SKETCH DENOTING STATION LOCATIONS

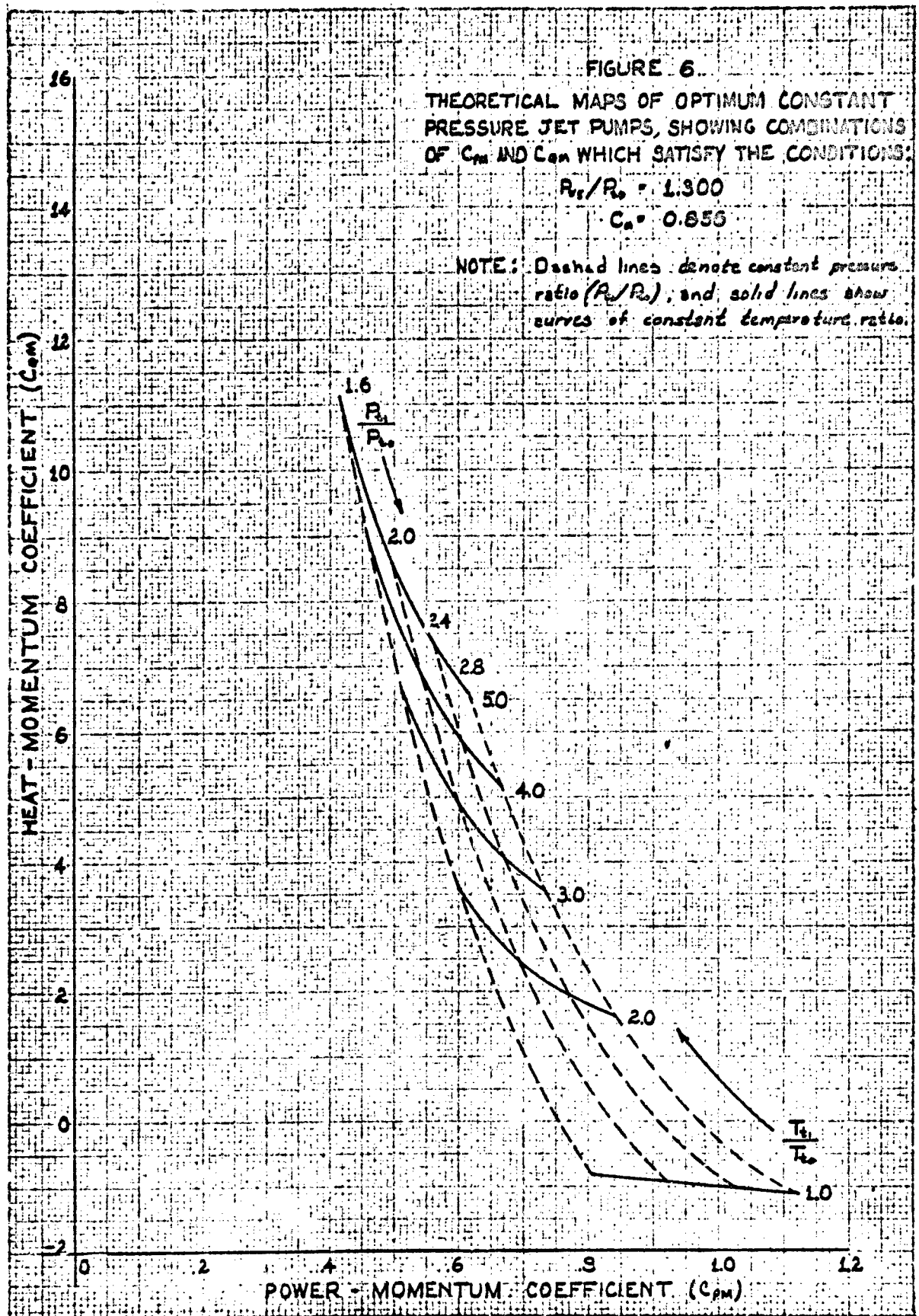


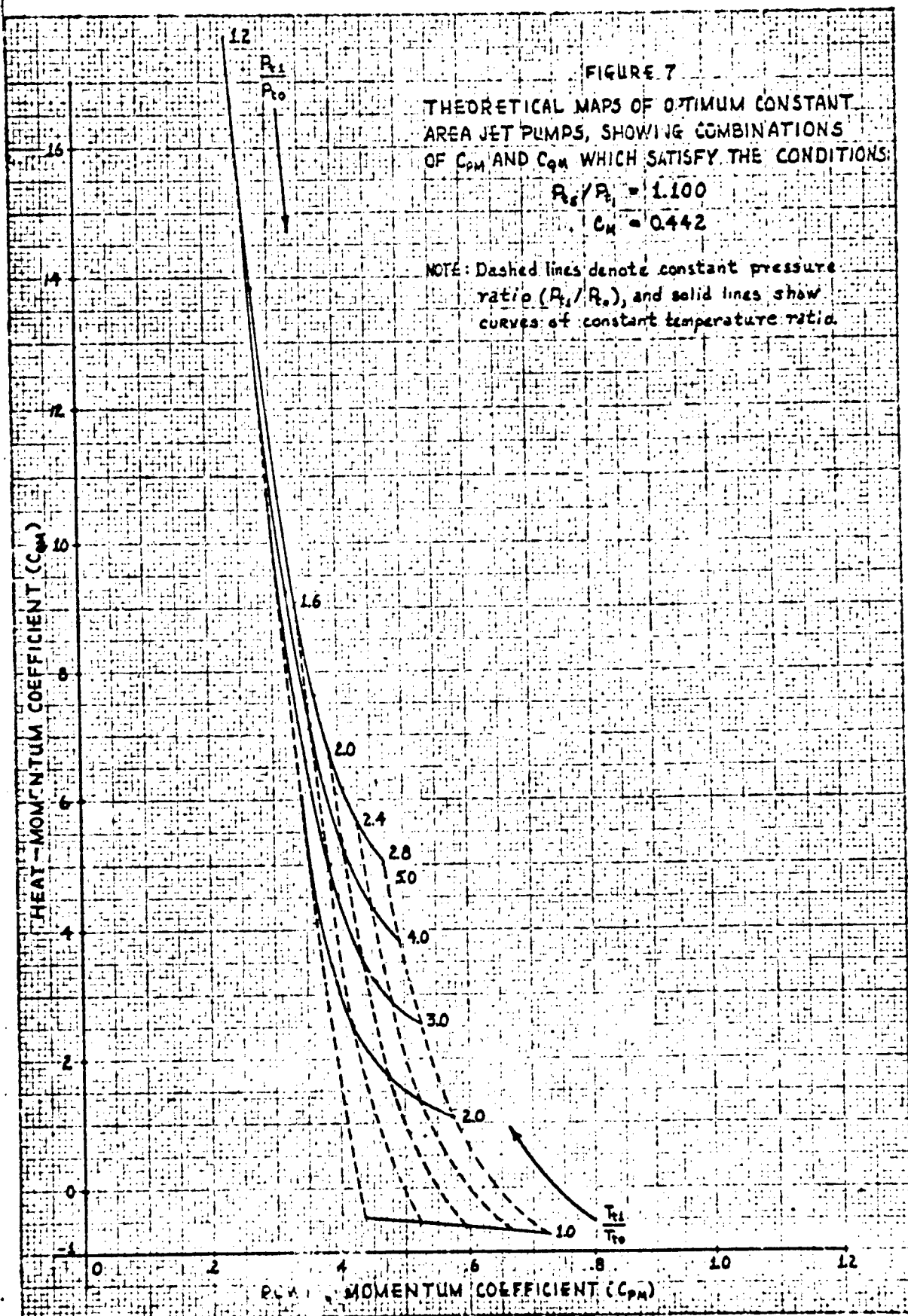


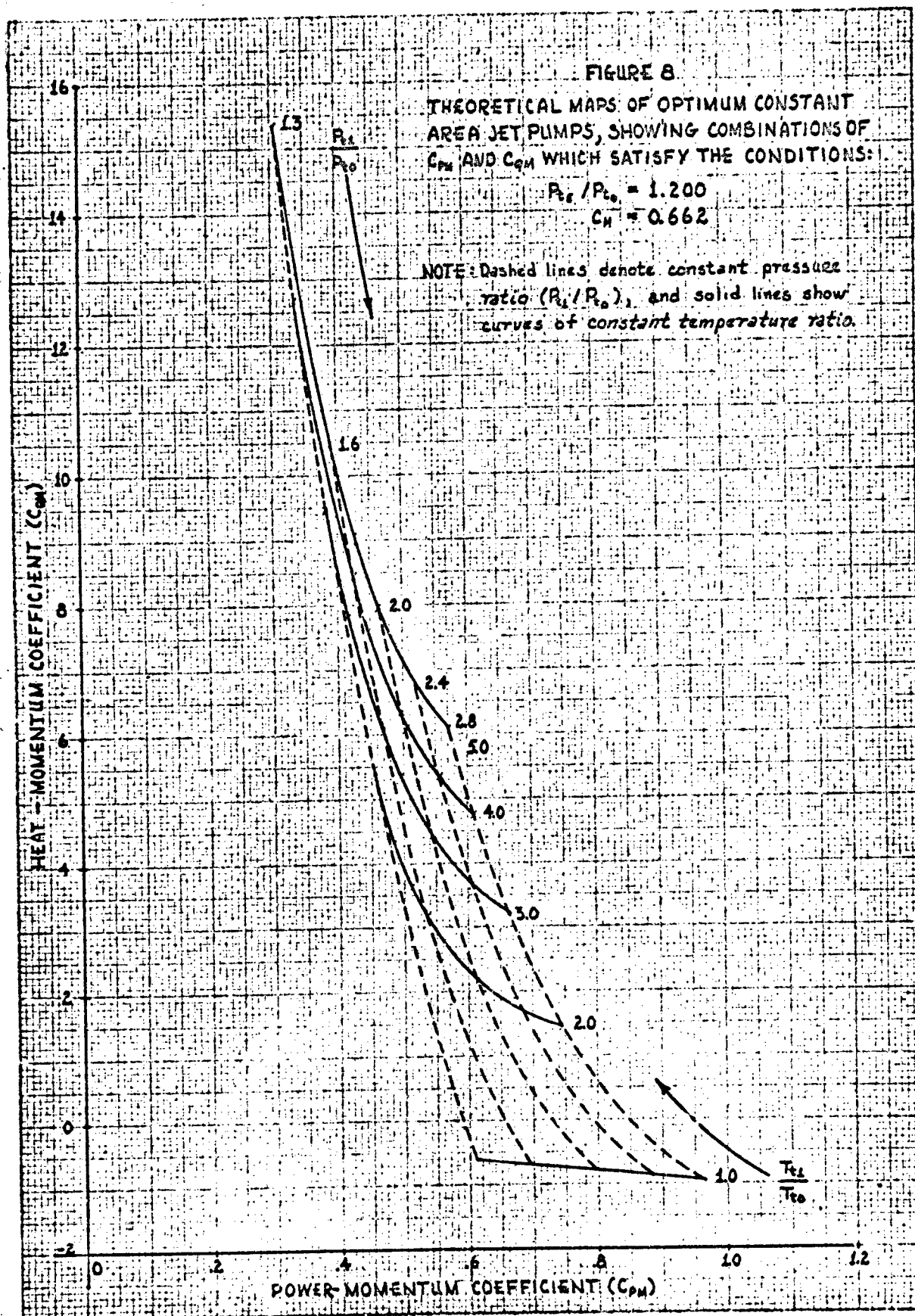


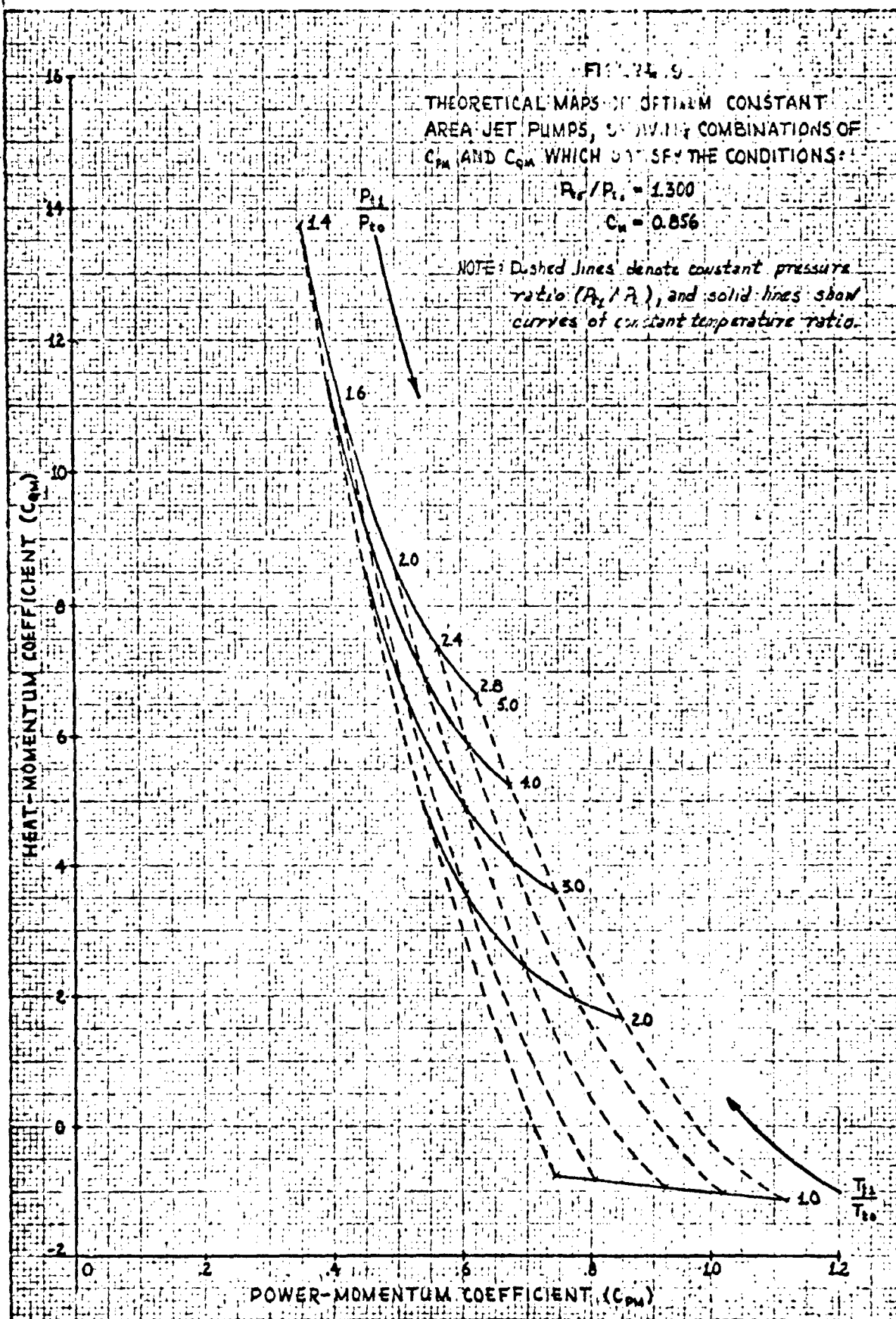


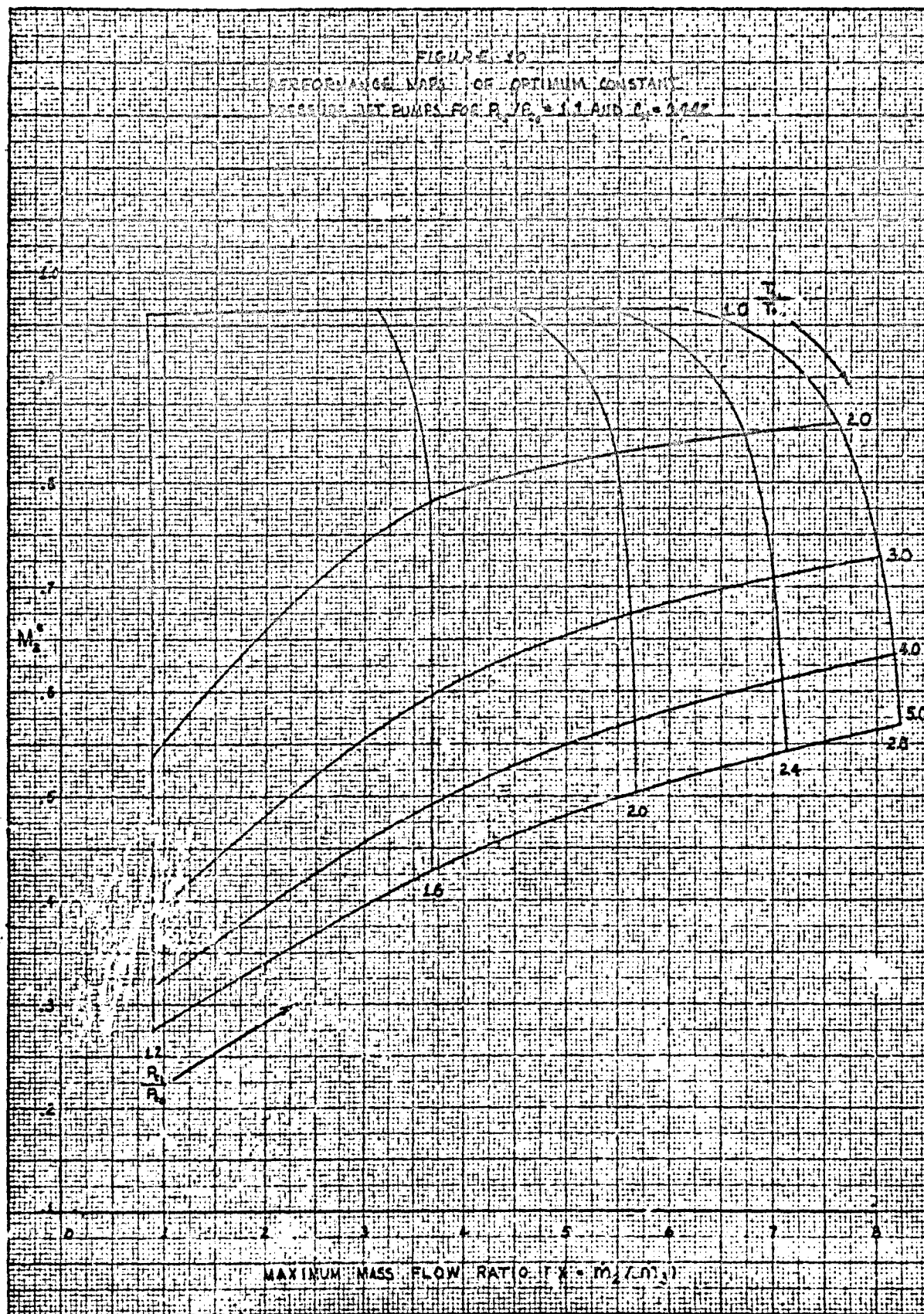


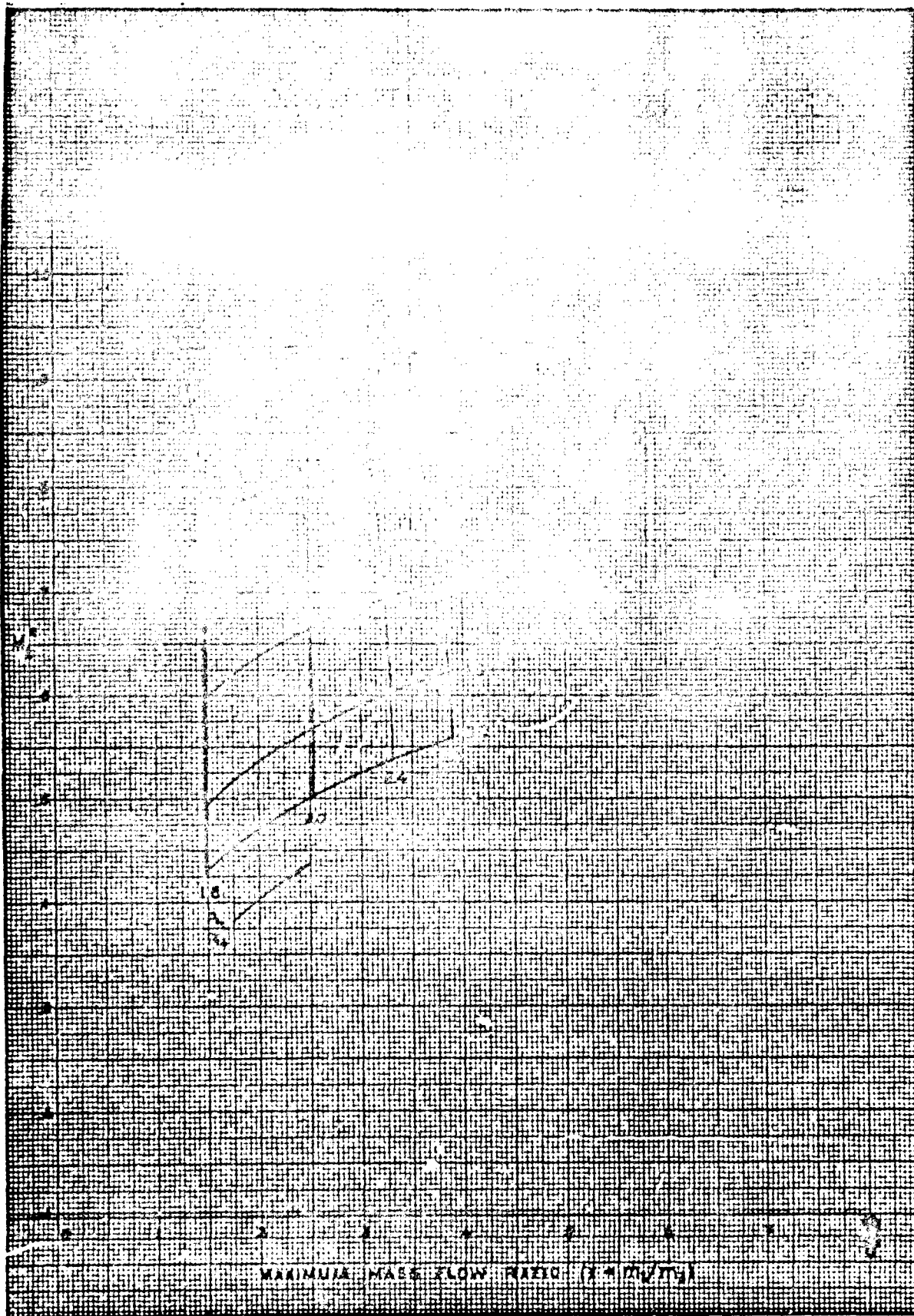


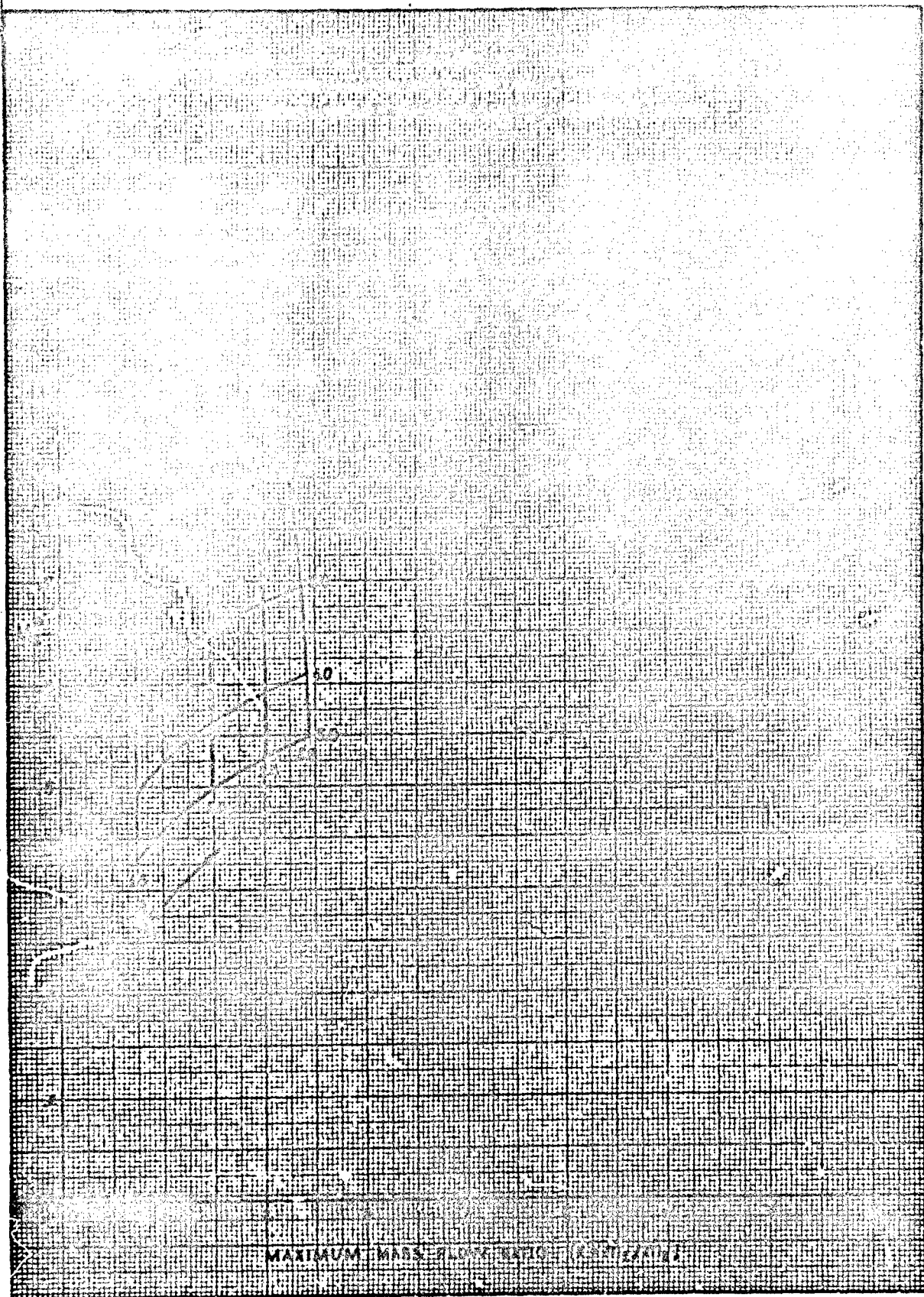


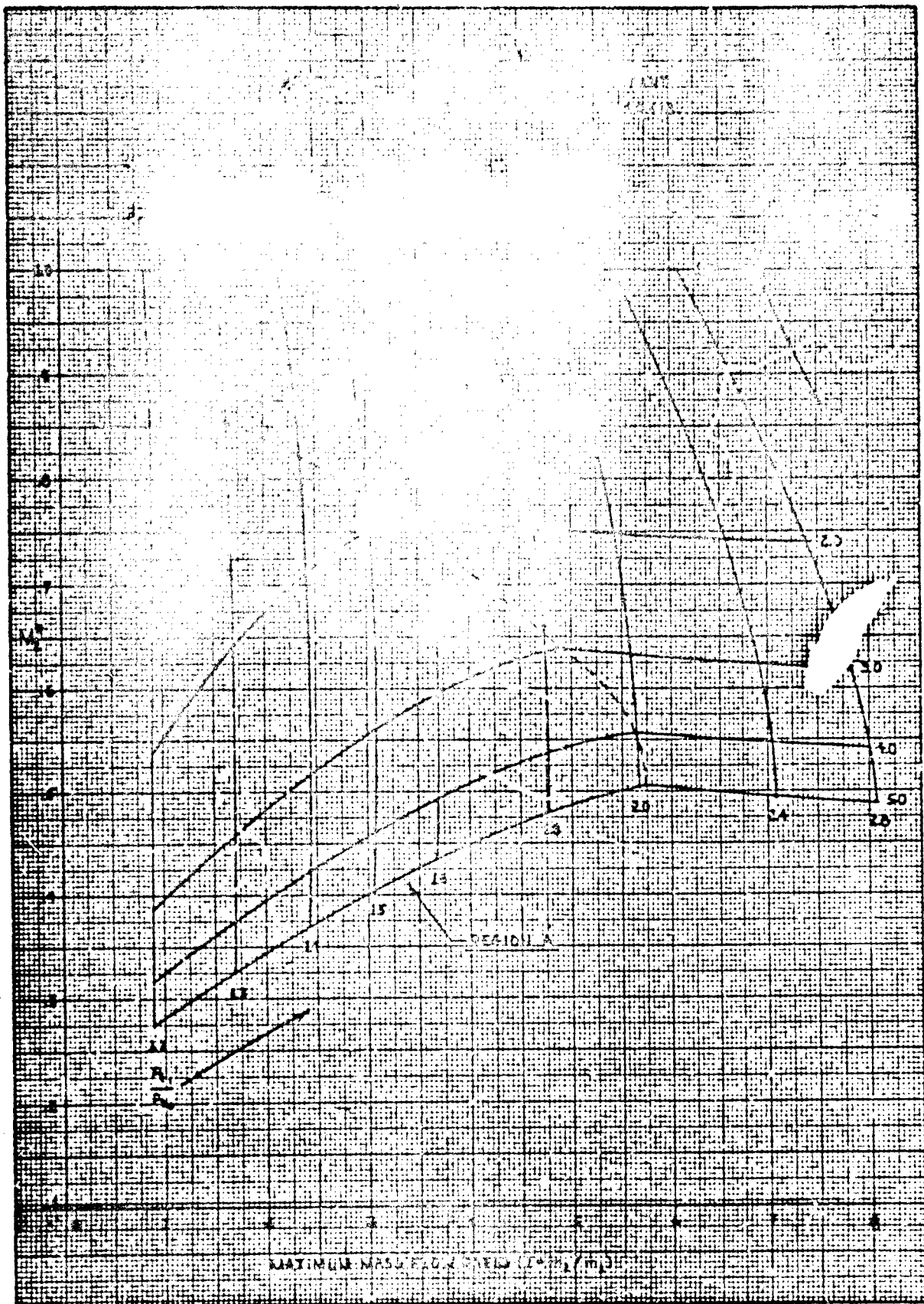


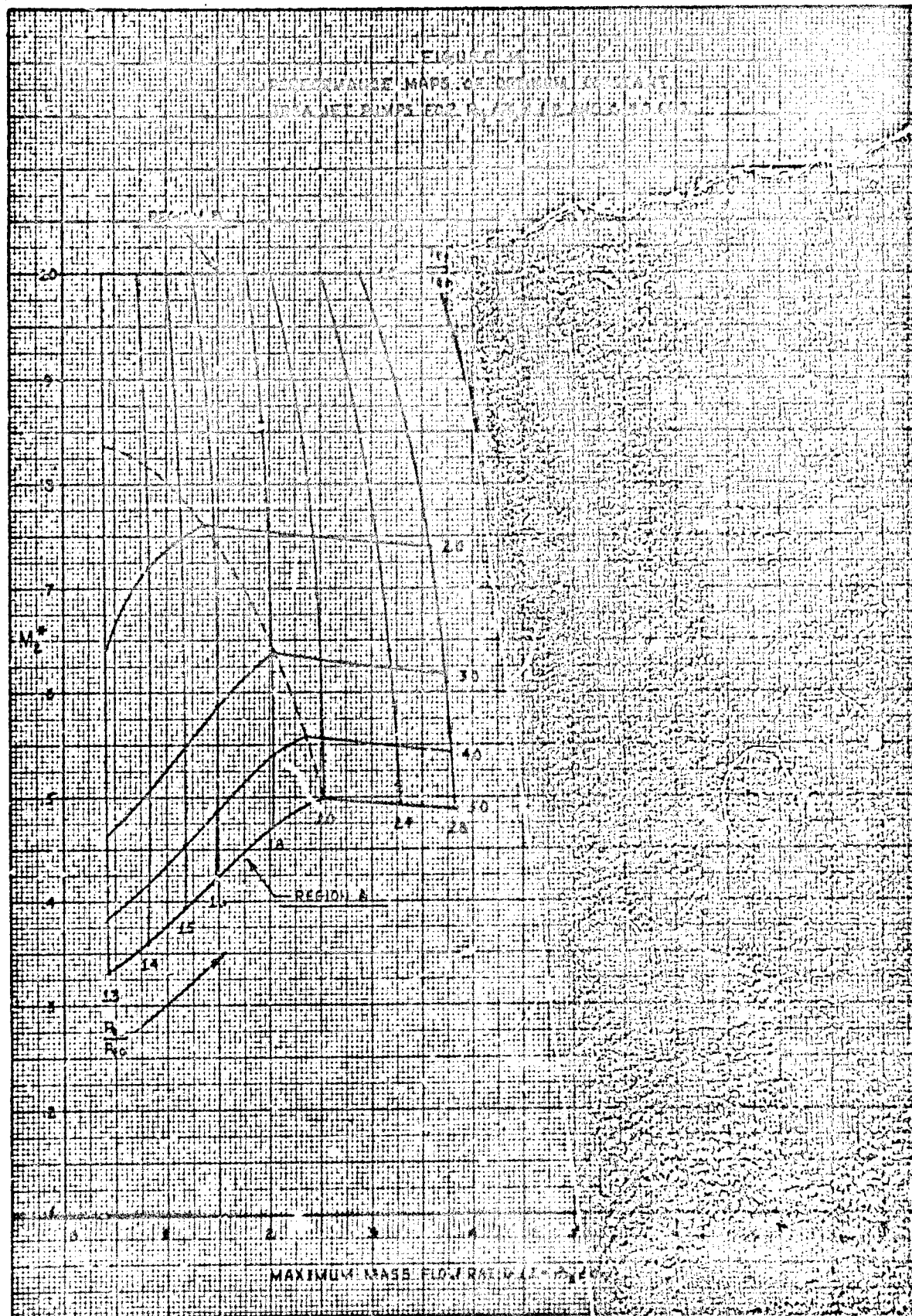


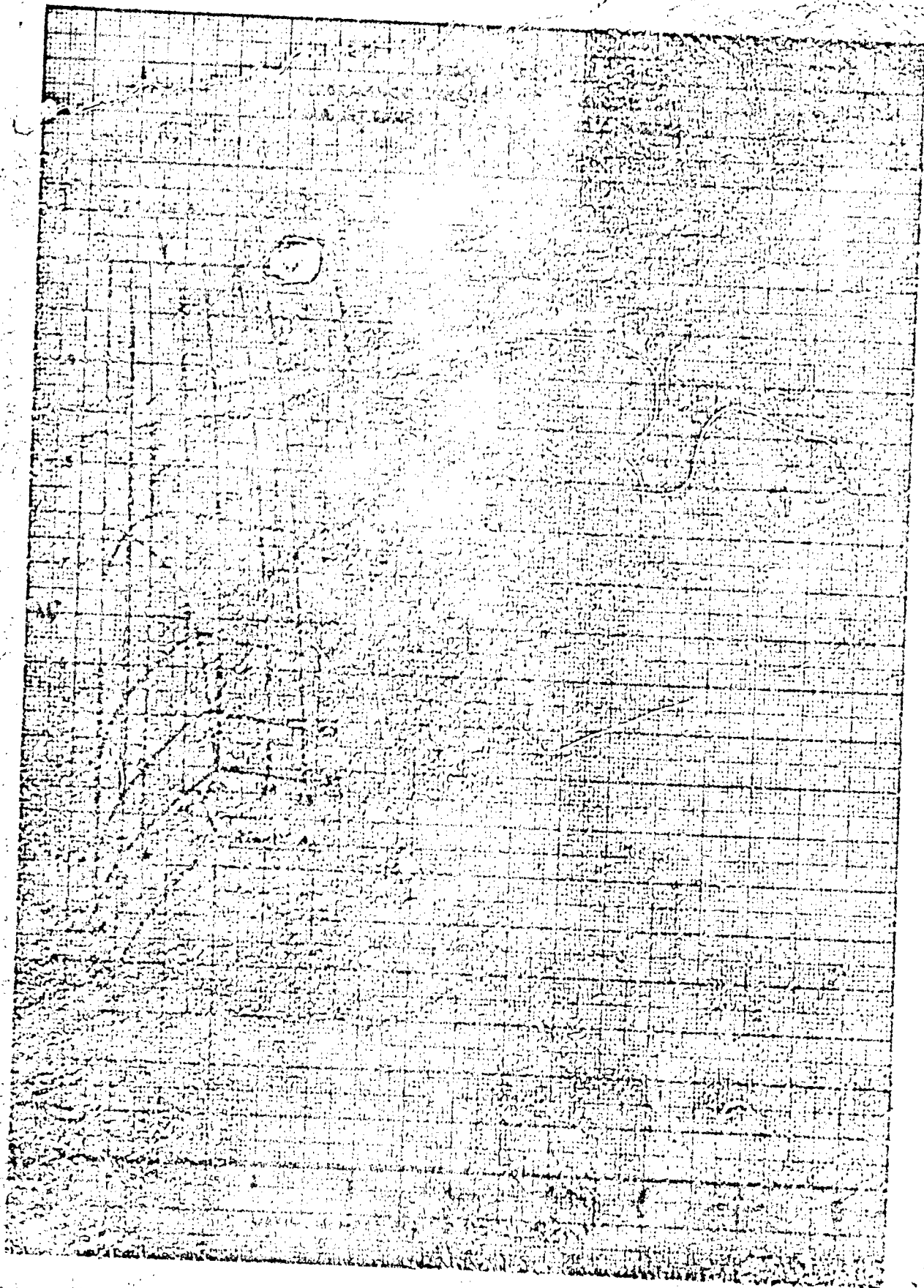


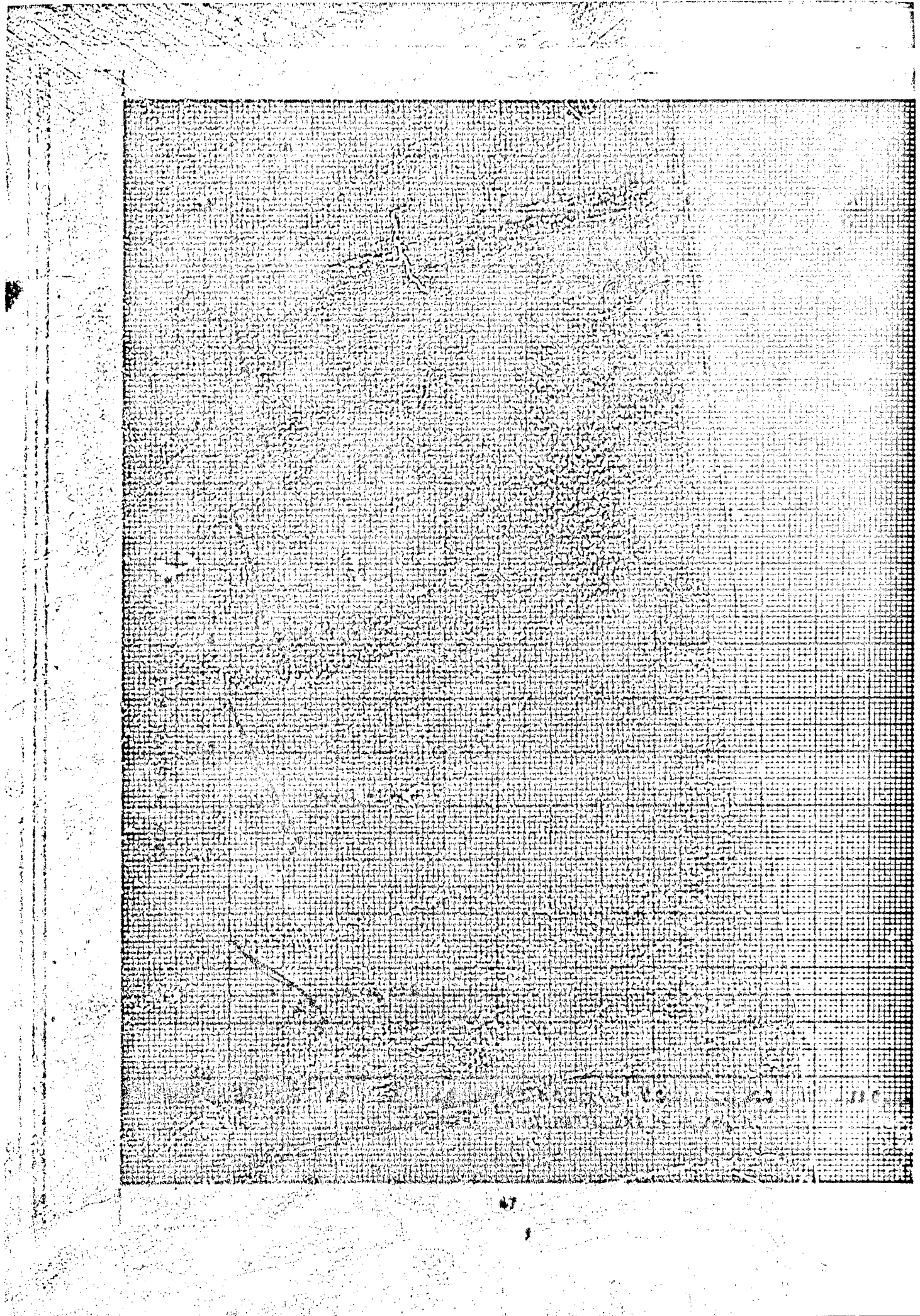


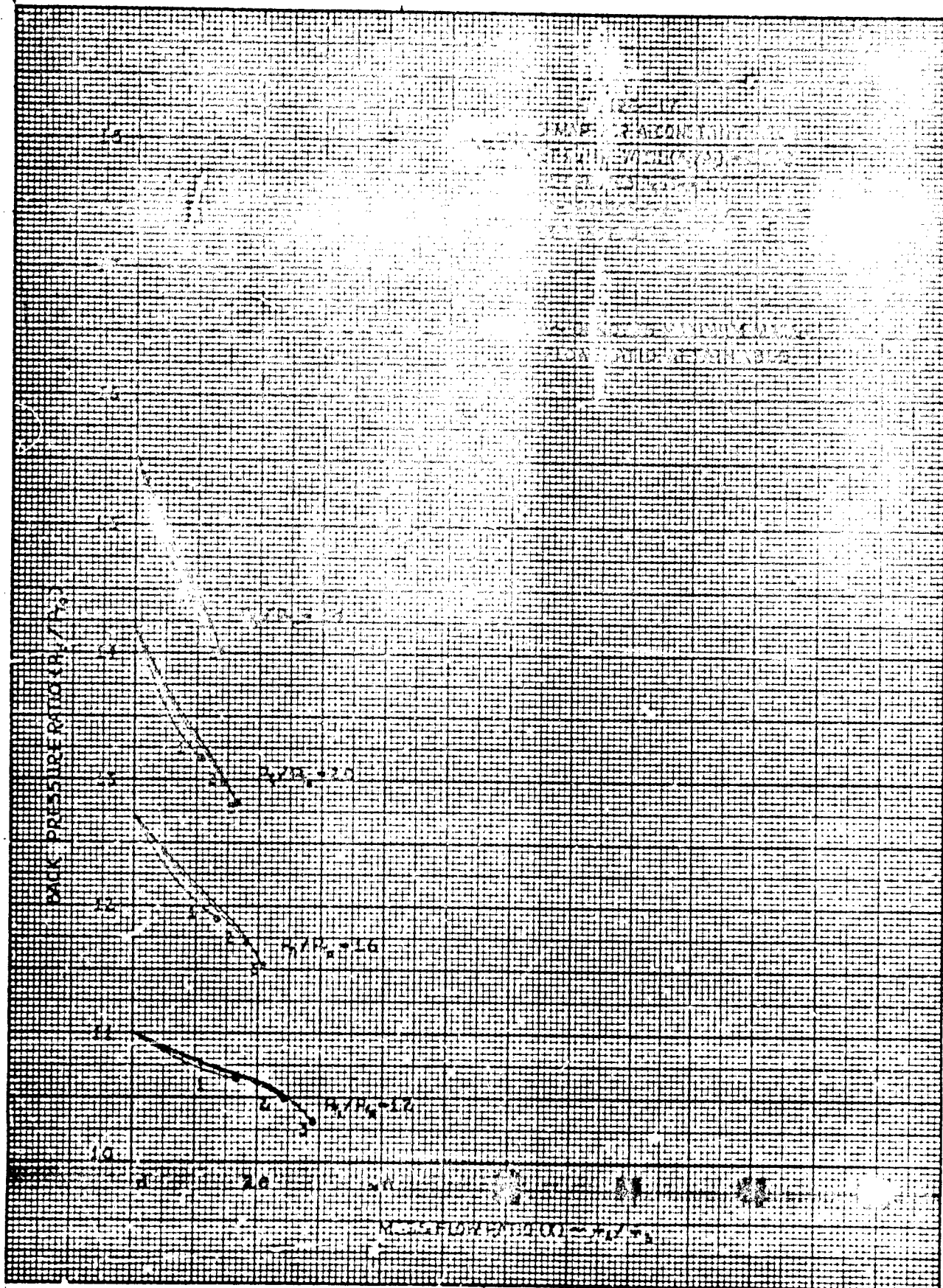


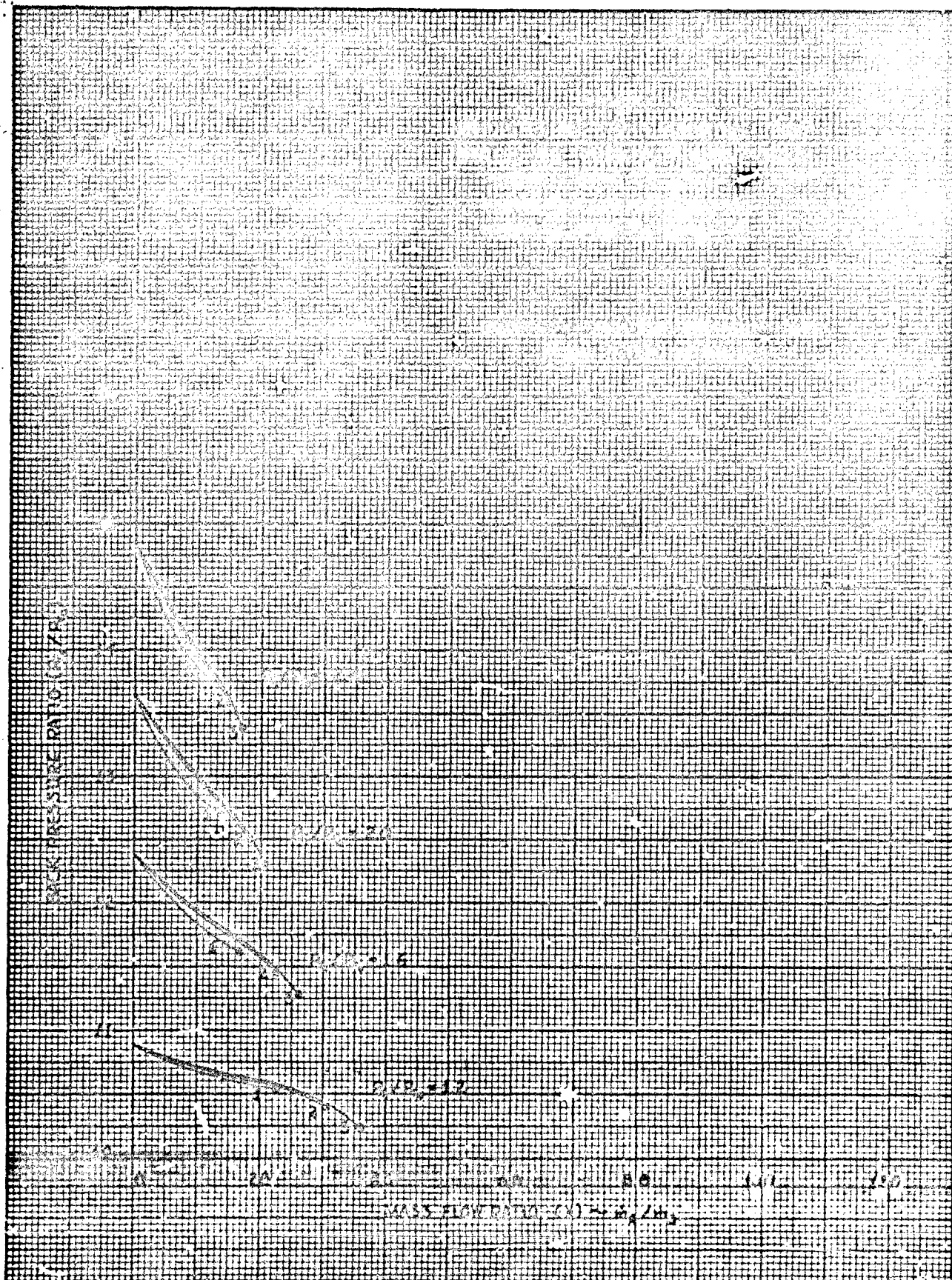


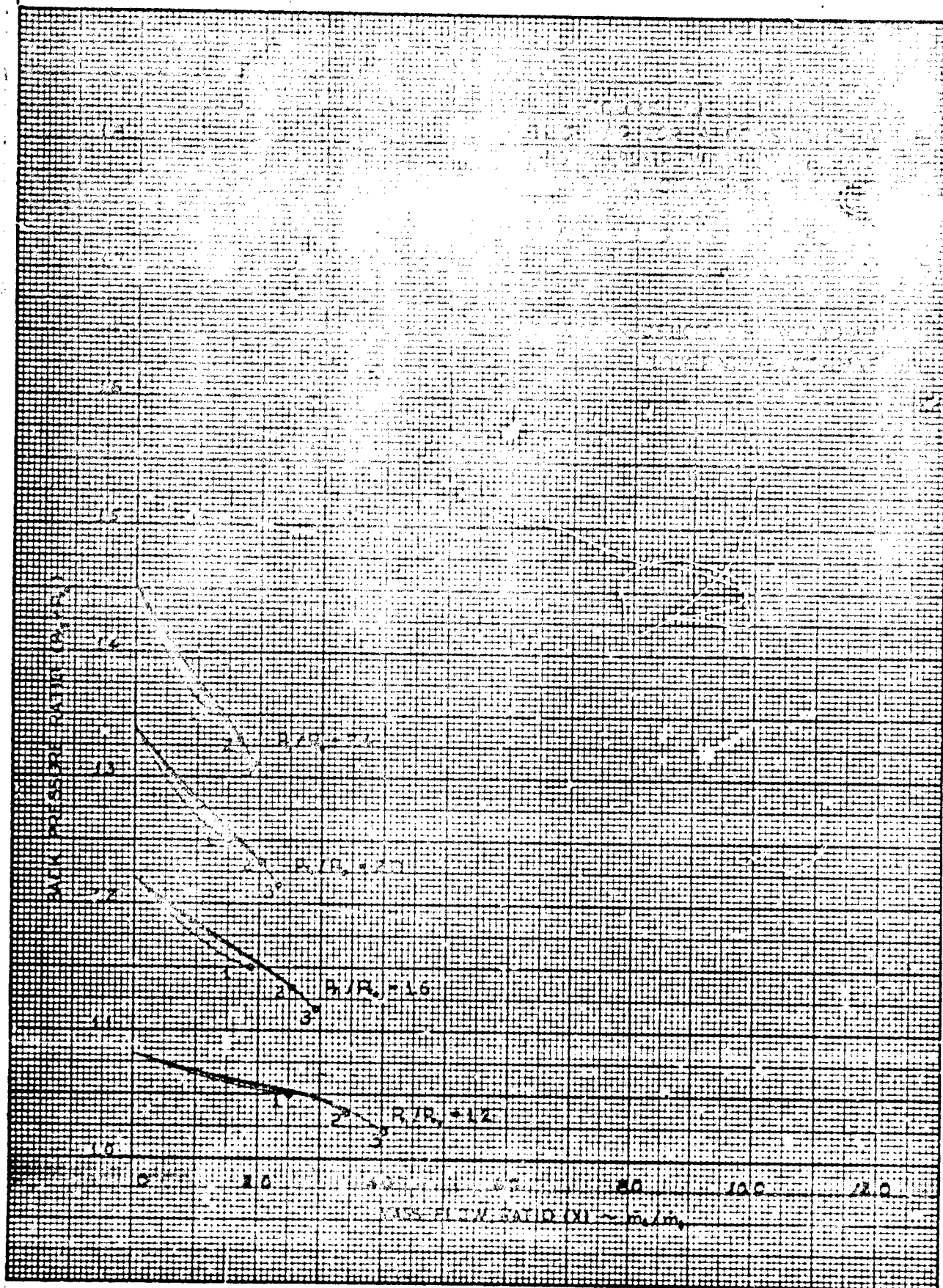


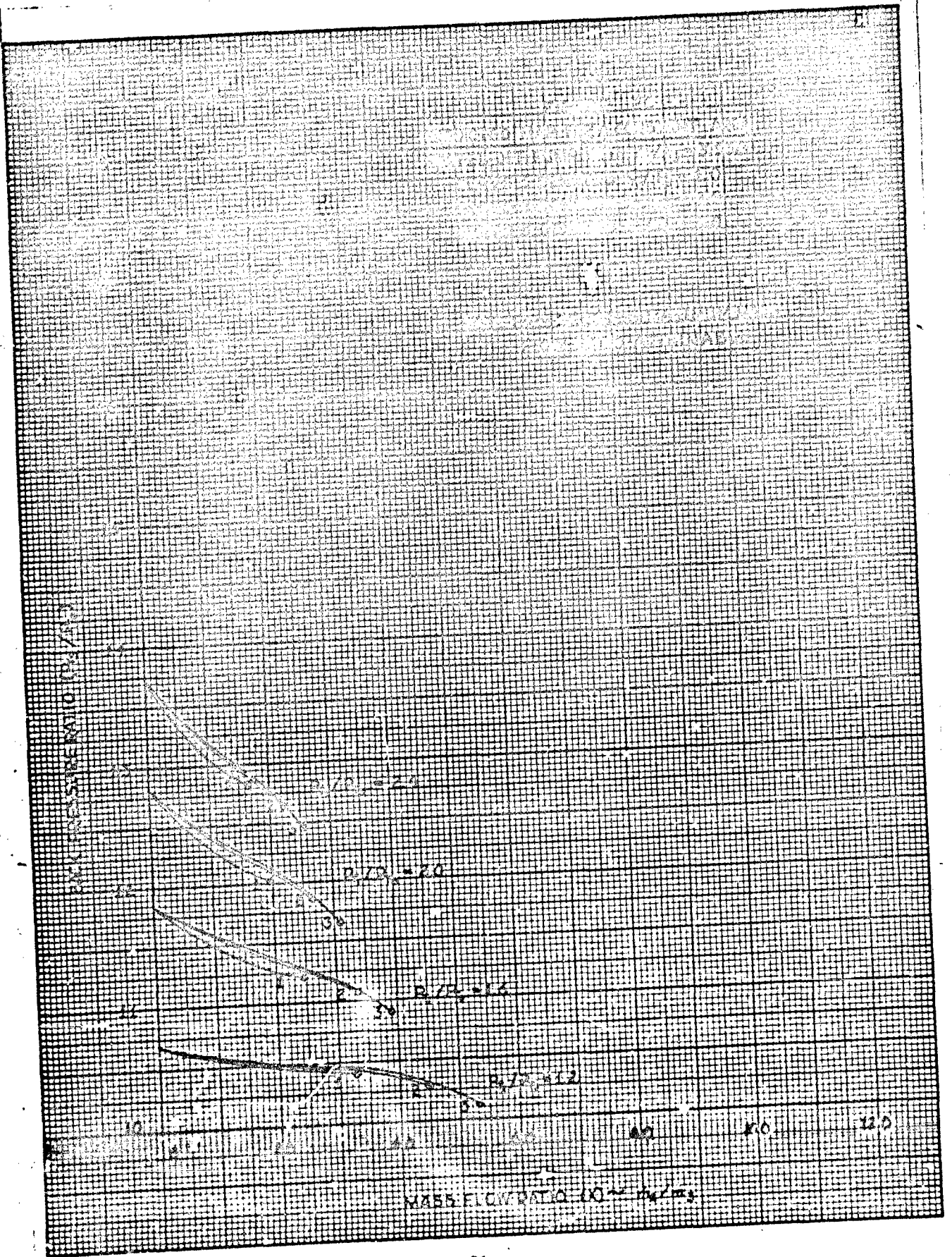


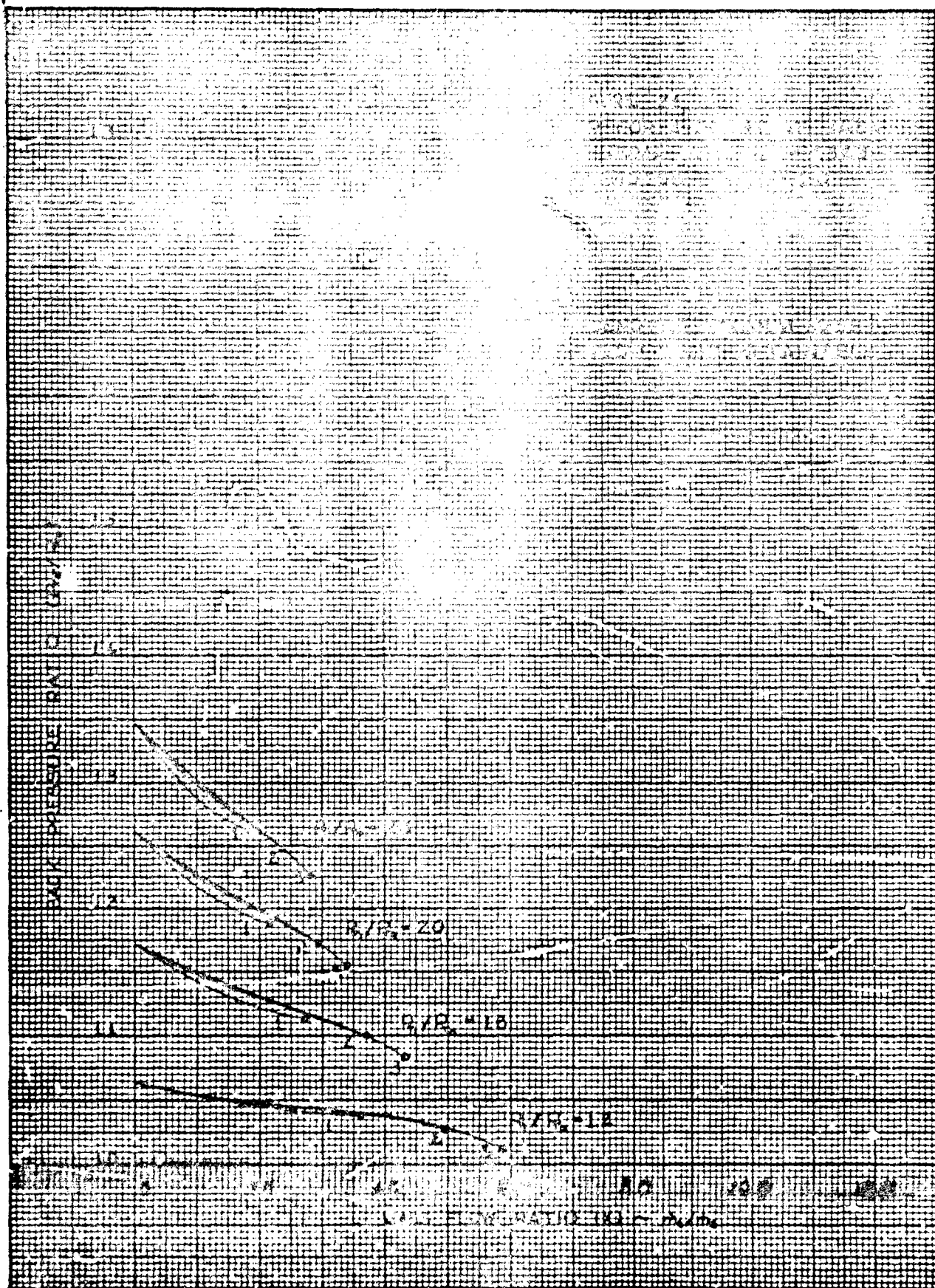




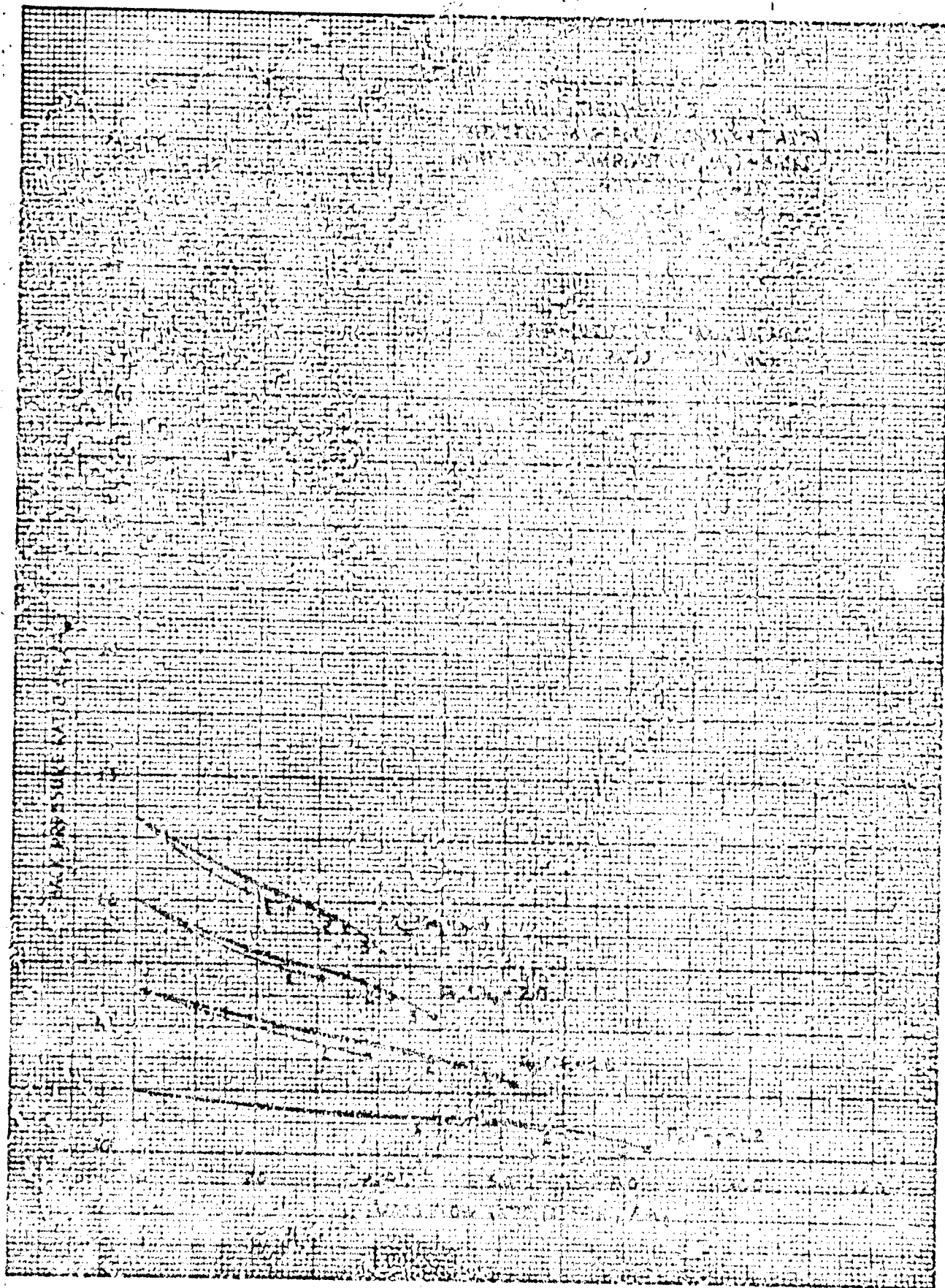












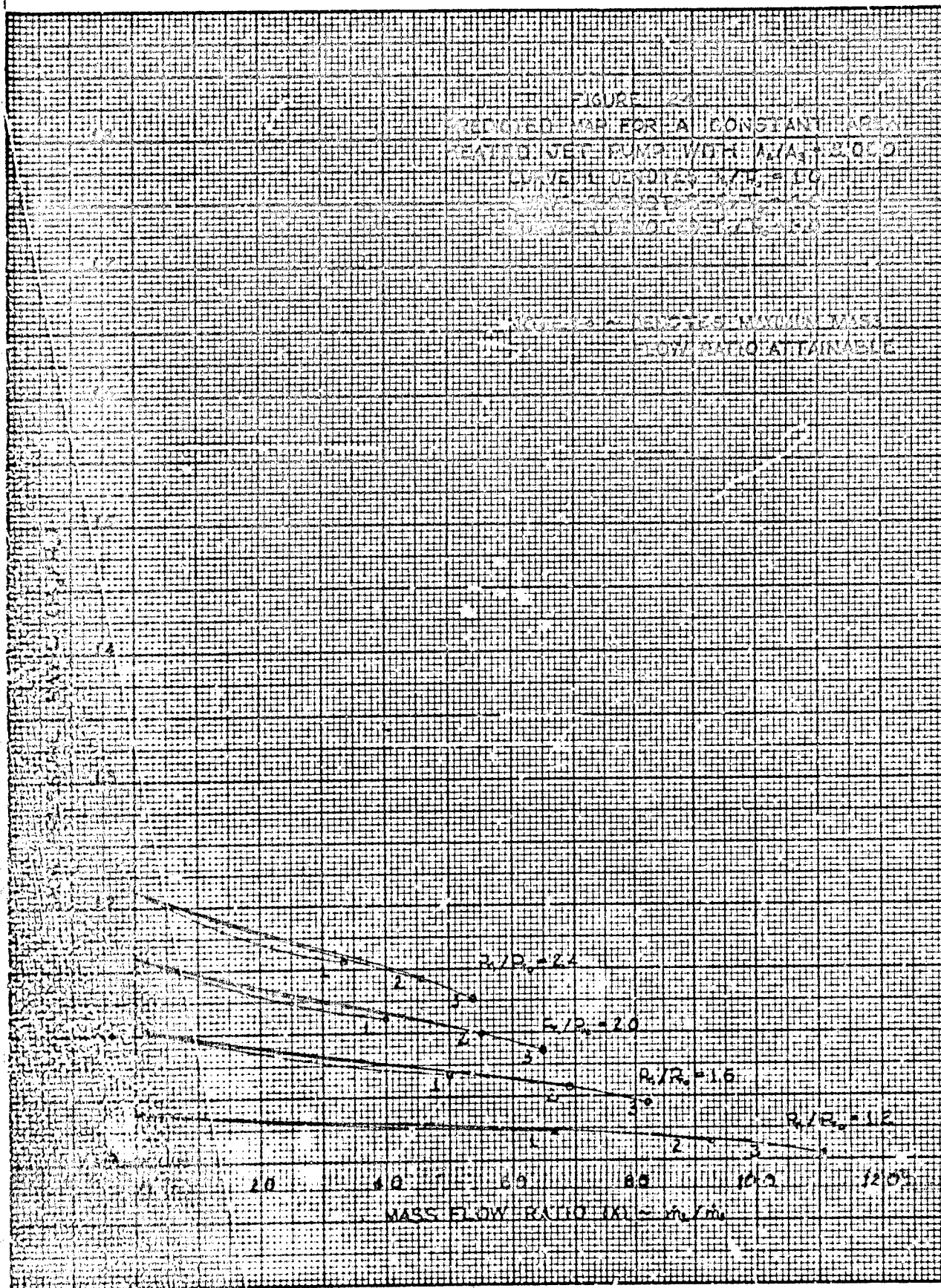


FIGURE 26	USNDG5, AERO. DEPT.
TWO INCH NOZZLE AND MIXING TUBE ENTRANCE	SCALE 2:1

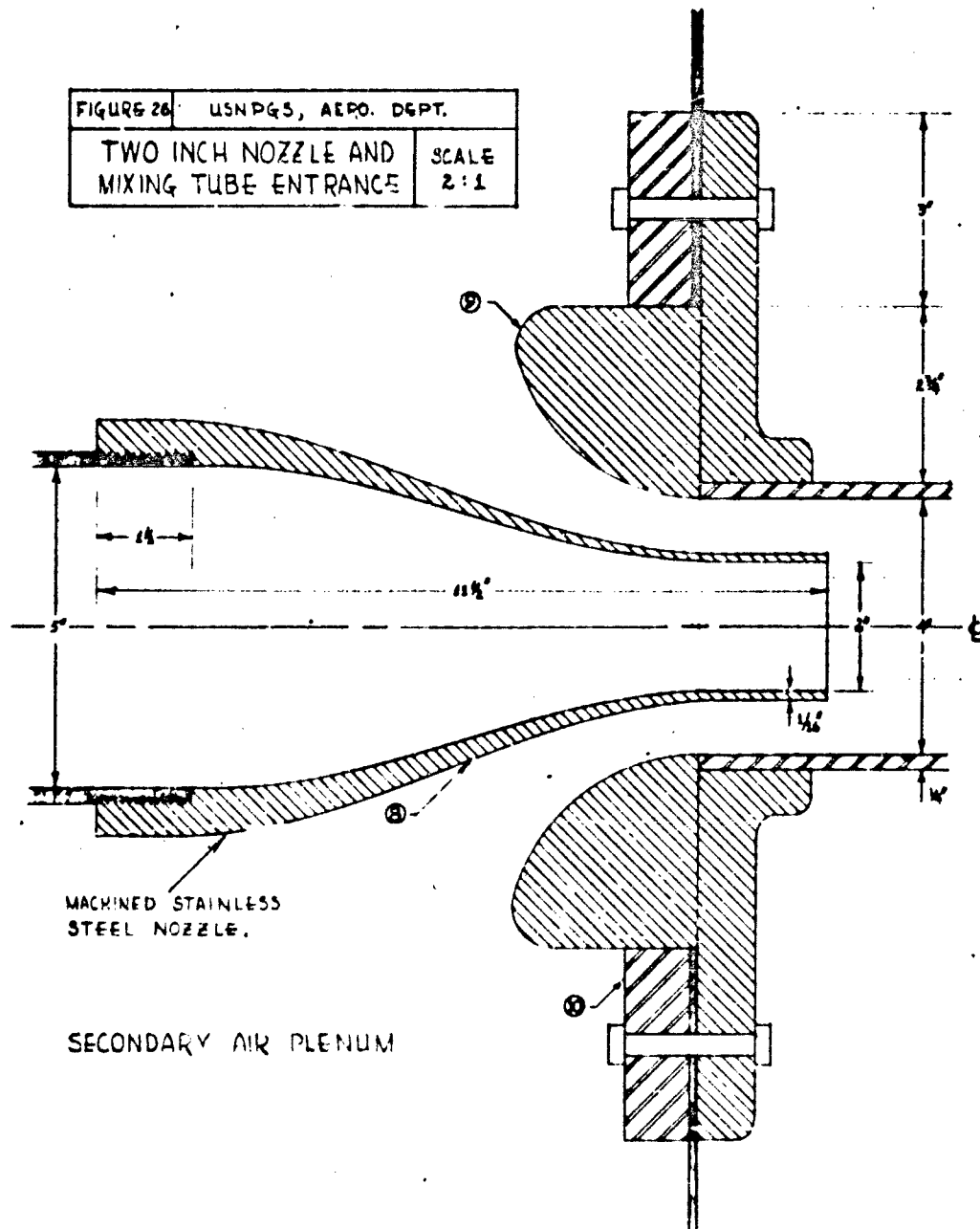


FIGURE 27 J. H. G. S. ARMO DEPT	
26 INCH NOZZLE AND MIXING TUBE ENTRANCE	SCALE 2:1

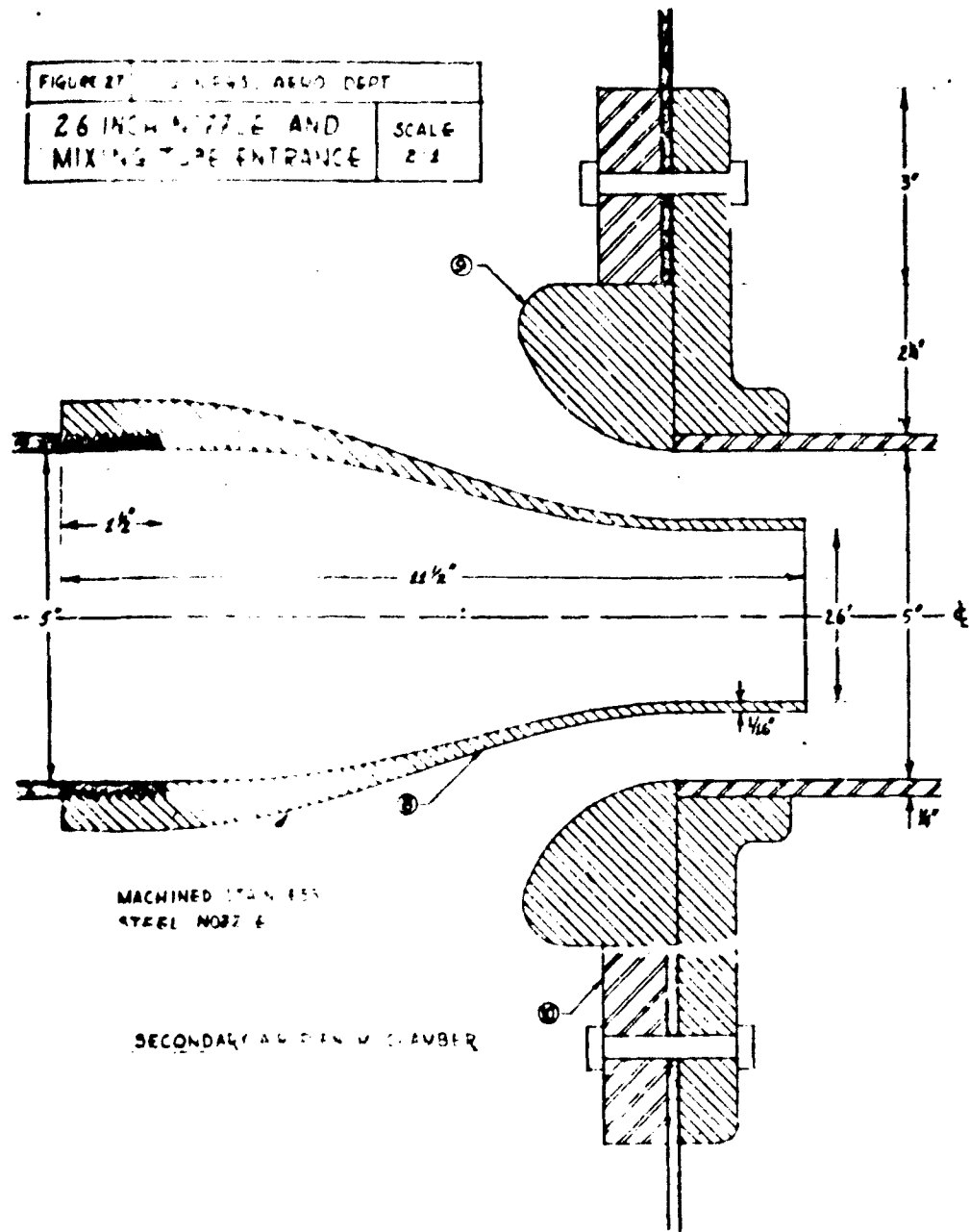
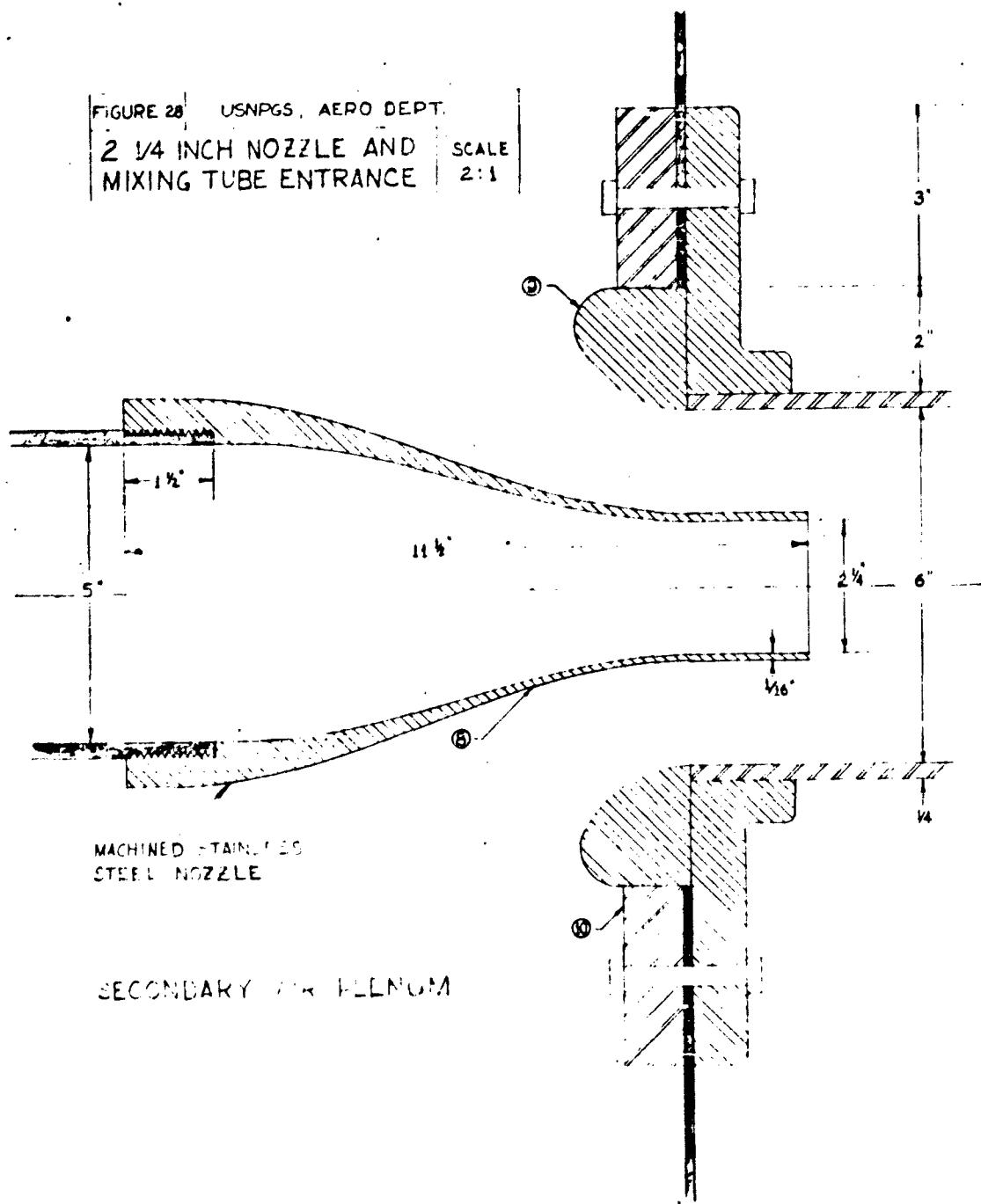


FIGURE 28 USNPGS, AERO DEPT.
2 1/4 INCH NOZZLE AND
MIXING TUBE ENTRANCE SCALE
2:1



RIDDER SUMMERS	USNPGS, DEPARTMENT OF AERONAUTICS	FIGURE 30
JAN 1966	MIXING TUBE STANCHIONS AND SUPPORT DESIGN	SCALE 8:1

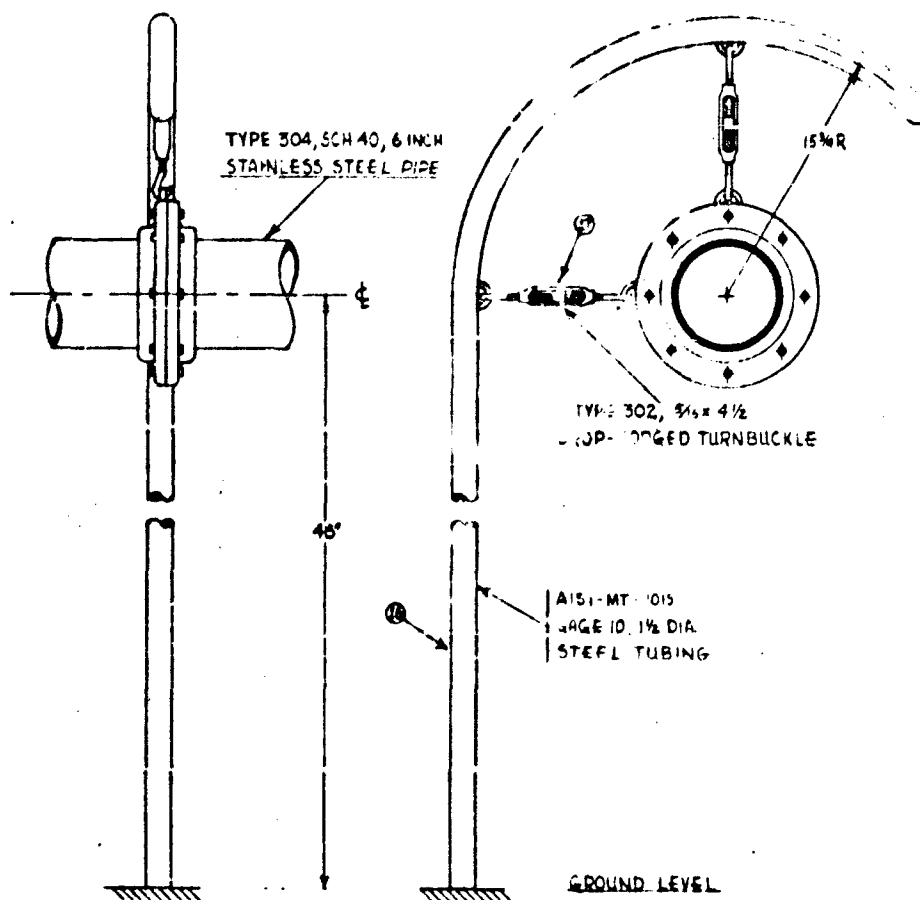
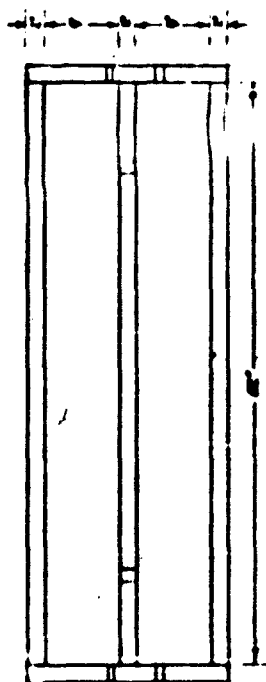
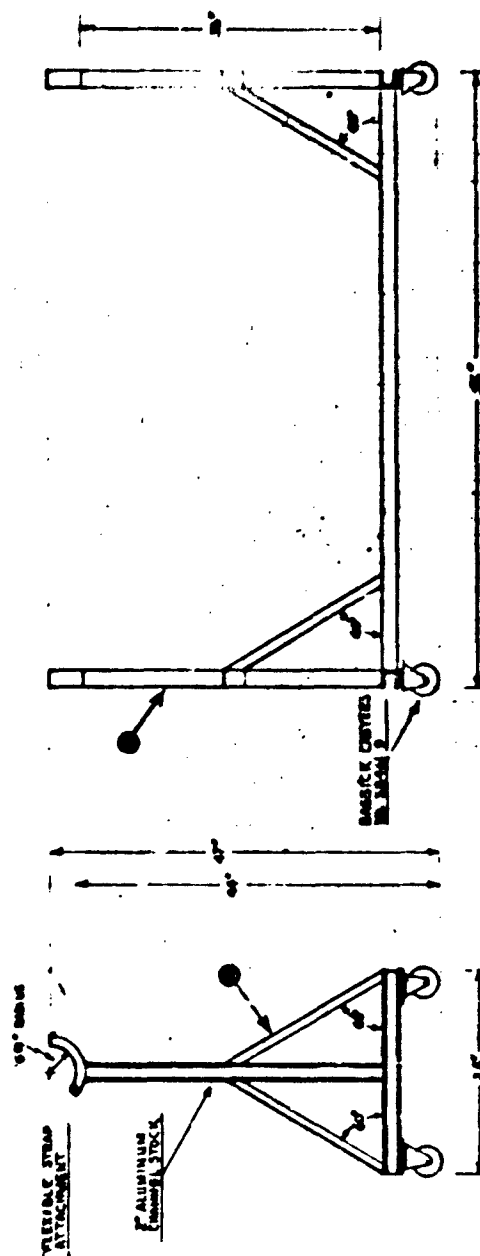


FIGURE 31	USNIPES, AERO DEPT.
MIXING TUBE HANDLING CART	SCALE 16:1



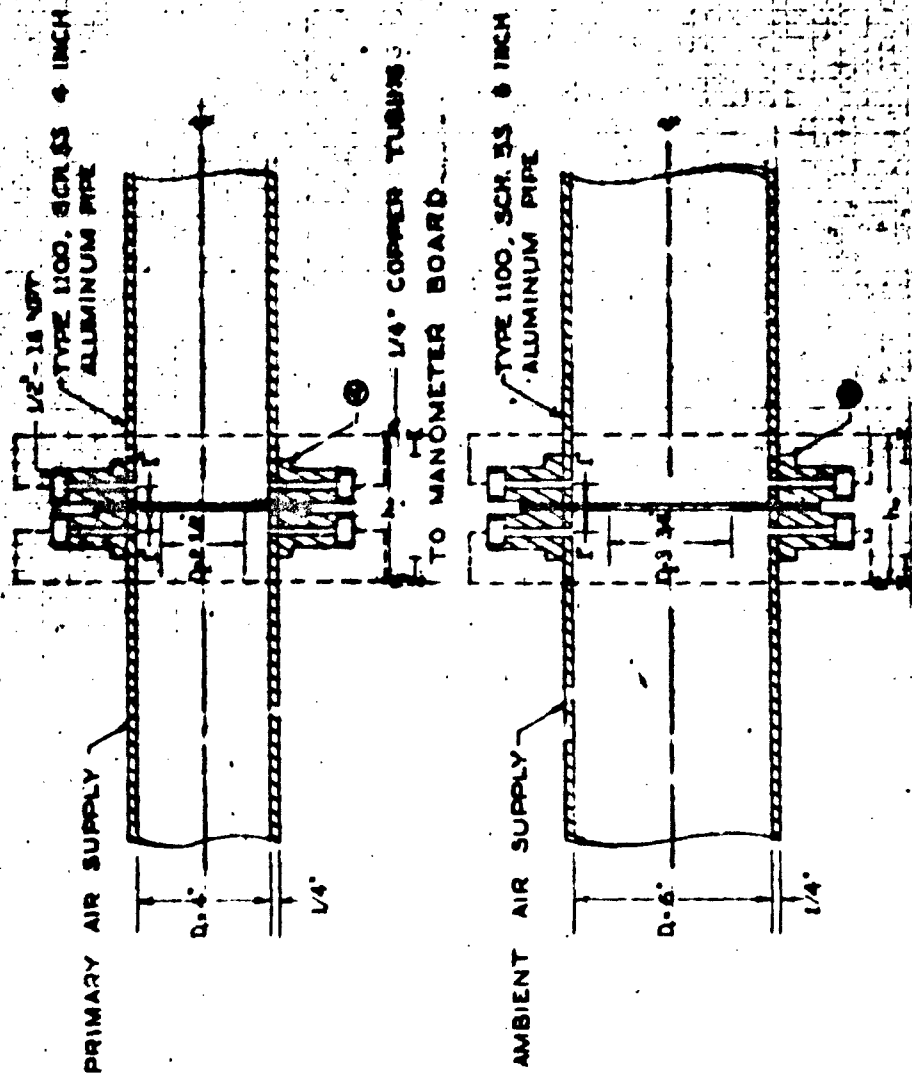
ALUMINUM CHANNEL AND BAR CONSTRUCTION
 MAXIMUM DESIGN LOAD 4000 LBS.
 MAXIMUM LOAD ANTICIPATED 300 LBS.



USNPGS, DEPARTMENT OF AERONAUTICS	FIGURE 32
MIXING TUBE INSTRUMENTATION	SCALE Z:1



SKETCH	USNPGS, DEPARTMENT OF AERONAUTICS	FIG. 1
REVISIONS	MASS FLOW	EC-11
JAN 1948	MEASUREMENT DESIGN	211



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APPENDIX A

Detailed Development

1. Preliminary relationships

The preliminary relationships are similar to those of Reference

2. All velocities are arbitrarily expressed in terms of the dimensionless velocity ratio, M^* . This ratio is obtained by dividing the flow velocity, V , by the reference velocity, a^* , which is the speed of sound at sonic velocity. The Mach number, M , is equal to M^* at sonic velocity and in all cases $aM = a^*M^*$. M^* is also proportional to a finite velocity at all absolute temperatures.

The following equations are used throughout the development:

$$a = \sqrt{\gamma g R T}$$

$$a^* = \sqrt{\frac{2\gamma}{\gamma+1} g R T^*}$$

$$\frac{1}{M^*} = 1 - \frac{\gamma-1}{\gamma+1} \dots$$

$$\frac{P}{P^*} = \left(\frac{T}{T^*} \right)^{\frac{\gamma}{\gamma-1}}$$

$$\frac{P}{P^*} = \left(\frac{T}{T^*} \right)^{\frac{1}{\gamma-1}}$$

$$\dot{m}_2 = \rho_2 A_2 V_2$$

$$\dot{m}_3 = \rho_3 A_3 V_3$$

$$C_p = \frac{R}{J} \frac{\gamma}{\gamma-1}$$

$$\gamma = \frac{C_p}{C_v}$$

$$p = \rho g R T$$

2. Performance coefficients

a. Momentum coefficient

The momentum coefficient is defined as the dimensionless momentum:

$$C_M = \frac{\rho_s A_s V_s^2}{\rho_o^* A_o a_o^{*2}} \quad (A-1)$$

Substituting $a_s^{*2} M_s^{*2}$ for V_s^2

$$C_M = \frac{\rho_s A_s a_s^{*2} M_s^{*2}}{\rho_o^* A_o a_o^{*2}} \quad (A-2)$$

Now $\rho_s a_s^{*2} M_s^{*2}$ can be written

$$\rho_s a_s^{*2} M_s^{*2} = \frac{\rho_s}{\rho_s^*} \rho_s^* a_s^{*2} M_s^{*2} \quad (A-3)$$

where

$$\frac{\rho_s}{\rho_s^*} = \left(\frac{T_s}{T_s^*} \right)^{\frac{1}{\gamma-1}}$$

which can be written

$$\frac{\rho_s}{\rho_s^*} = \left(\frac{T_s/T_{ts}}{T_s^*/T_{ts}} \right)^{\frac{1}{\gamma-1}} \quad (A-4)$$

Substituting for T_s/T_{ts} and T_s^*/T_{ts} into (A-4)

$$\frac{\rho_s}{\rho_s^*} = \left(\frac{1 - \frac{\gamma-1}{\gamma+1} M_s^{*2}}{\frac{2}{\gamma+1}} \right)^{\frac{1}{\gamma-1}}$$

$\rho_s^* a_s^{*2}$ can be written in terms of the equation of state and the speed of sound at sonic velocity.

$$\rho_s^* a_s^{*2} = \left(\frac{p_s^*}{\gamma R T_s^*} \right) (\gamma g R T_s^*)$$

which can be written

$$\rho_s^* a_s^{*2} = \gamma \left(\frac{p_s^*}{R T_s^*} \right) p_{ts}$$

Since p_s^* corresponds to static pressure at $M_s = M_s^* = 1$, then,

$$\rho_s^* a_s^{*2} = \gamma \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}} p_{ts}$$

Substituting for ρ_s/ρ_s^* and $\rho_s^* a_s^{*2}$ into (A-3)

$$\rho_s a_s^{*2} M_s^{*2} = \left(\frac{1 - \frac{\gamma-1}{\gamma+1} M_s^{*2}}{\frac{2}{\gamma+1}} \right)^{\frac{1}{\gamma-1}} \gamma \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}} p_{ts} M_s^{*2}$$

Similar to $\rho_s^* a_s^{*2}$

$$\rho_o^* a_o^{*2} = \gamma \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}} p_{to}$$

Dividing $\rho_s a_s^{*2} M_s^{*2}$ by $\rho_o^* a_o^{*2}$

$$\frac{\rho_s a_s^{*2} M_s^{*2}}{\rho_o^* a_o^{*2}} = \left(\frac{1 - \frac{\gamma-1}{\gamma+1} M_s^{*2}}{\frac{2}{\gamma+1}} \right)^{\frac{1}{\gamma-1}} \frac{p_{ts}}{p_{to}} M_s^{*2}$$

From continuity and remembering $M_o^* = 1$

$$\rho_s A_s V_s = \rho_o A_o V_o$$

$$\frac{A_s}{A_o} = \frac{\rho_o V_o}{\rho_s V_s}$$

Substituting for V_5 and V_6

$$\frac{A_5}{A_6} = \frac{P_6 M_6^* a_6^*}{P_5 M_5^* a_5^*} \quad (A-5)$$

Since $T_{t6} = T_{t5}$, therefore $a_6^* = a_5^*$, hence

$$\frac{A_5}{A_6} = \frac{P_6}{P_5 M_5^*}$$

Substituting for $(P_6/P_{t6})/(P_5/P_{t5})$ in (A-5) and noting $P_{t6} = P_{t5}$

$$\frac{A_5}{A_6} = \left(\frac{1 - \frac{\gamma}{\gamma+1} M_5^{*2}}{1 - \frac{\gamma-1}{\gamma+1} M_5^{*2}} \right)^{\frac{\gamma}{\gamma-1}} \frac{1}{M_5^*}$$

Substituting into Equation (A-2) for

$$\frac{P_6 a_6^* M_6^*}{P_5^* a_5^* M_5^*}$$

and A_5/A_6 , the result is

$$C_M = \frac{P_{t5}}{P_{t0}} M_5^*$$

and substituting for M_5^*

$$C_M = \frac{P_{t5}}{P_{t0}} \left\{ \frac{\gamma+1}{\gamma-1} \left[1 - \left(\frac{P_{t0}}{P_{t5}} \right)^{\frac{\gamma-1}{\gamma}} \right] \right\}^{\frac{1}{2}}$$

It is seen that C_M is a function only of the required discharge total pressure ratio. In particular, C_M is independent of the amount of heat supplied. By specifying the desired output pressure ratio, P_{t5}/P_{t0} , the required C_M is determined.

b. Power-momentum coefficient

The ratio of the power coefficient to the momentum coefficient defines the power-momentum coefficient.

$$C_{PM} = \frac{C_P}{C_M} = \frac{P_5 A_5 V_5 a_5^* c_p \Delta T_c}{P_5 A_5 V_5^2 \frac{a_5^{*2}}{2gJ}}$$

$$C_{PM} = \frac{\dot{m}_2}{\dot{m}_2 + \dot{m}_3} \frac{\Delta H_c 2gJ}{V_5 a_5^*}$$

$$C_{PM} = \frac{\Delta H_c \cdot 2gJ}{\left(\frac{m_2}{m_1} + 1\right) V_s a_s^*} \quad (A-6)$$

$$\Delta H_c = c_p (T_{t1c} - T_{t0})$$

$$\Delta H_c = c_p T_{t0} \left\{ \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}$$

Substituting for ΔH_c and V_s in (A-6), then C_{PM} becomes

$$C_{PM} = \frac{c_p T_{t0} \left\{ \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\} 2gJ}{\left(\frac{m_2}{m_1} + 1\right) a_s^* M_s^* a_s^*}$$

also

$$\frac{2gJ c_p T_{t0}}{a_s^* a_s^*} = \frac{2gJR \gamma T_{t0}}{\left(\frac{\gamma-1}{\gamma} g R T_{t0}\right)^{\frac{\gamma}{\gamma-1}} \left(\frac{\gamma-1}{\gamma} g R T_{t0}\right)^{\frac{\gamma}{\gamma-1}} J(\gamma-1)} \quad (A-7)$$

Therefore

$$C_{PM} = \frac{\left\{ \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\} \frac{\gamma+1}{\gamma-1}}{\left(\frac{m_2}{m_1} + 1\right) \sqrt{\frac{T_{t1}}{T_{t0}}} M_s^*} \quad (A-8)$$

rearranging and substituting in (A-8) for T_{t1}/T_{t0} from the energy equation (A-31)

$$\frac{C_{PM}}{\left\{ \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}} = \frac{\frac{\gamma+1}{\gamma-1} (M_s^*)}{\left\{ (\gamma+1) \left(\gamma + \frac{\gamma-1}{T_{t0}} \right) \right\}} \quad (A-9)$$

c. Heat-momentum coefficient

The ratio of the heat coefficient to the momentum coefficient

defines the heat-momentum coefficient.

$$C_{qm} = \frac{C_q}{C_m} = \frac{P_3 A_3 V_3 a_o^*}{P_5 A_5 V_5^2} \frac{c_p \Delta T_B}{\frac{a_o^{*2}}{2gJ}}$$

$$C_{qm} = \frac{\dot{m}_3}{\dot{m}_2 + \dot{m}_3} \frac{\Delta q \cdot 2gJ}{V_5 a_o^*}$$

$$C_{qm} = \frac{\Delta q}{(\gamma + 1)} \frac{2gJ}{a_o^* M_5^* a_o^*}$$

$$\Delta q = c_p (T_{t1} - T_{t1c})$$

$$\Delta q = c_p T_{t0} \left(\frac{T_{t1}}{T_{t0}} - \frac{T_{t1c}}{T_{t0}} \right)$$

Since there is an isentropic relationship between state 0 and 1c, then Δq becomes

$$\Delta q = c_p T_{t0} \left\{ \frac{T_{t1}}{T_{t0}} - \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} \right\}$$

Following a similar substitution for $\frac{2gJ c_p T_{t0}}{a_o^* a_o^*}$ as in (A-7) C_{qm} becomes:

$$C_{qm} = \frac{\left\{ \frac{T_{t1}}{T_{t0}} - \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} \right\} \frac{\gamma+1}{\gamma-1}}{(\gamma+1) \sqrt{\frac{T_{t1}}{T_{t0}}} M_5^*} \quad (A-10)$$

Rearranging and substituting for T_{t1}/T_{t0} from the energy equation (A-31)

$$\frac{C_{qm}}{\left\{ \frac{T_{t1}}{T_{t0}} - \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} \right\}} = \frac{\frac{\gamma+1}{\gamma-1} (M_5^*)^{-1}}{\left\{ (\gamma+1) \left(\gamma + \frac{T_{t1}}{T_{t0}} \right) \right\}^{1/2}} \quad (A-11)$$

It is noted that the following equation can be written involving C_{qm} and C_{pm} from (A-9) and (A-11).

$$\frac{C_{pm}}{\left\{ \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}} = \frac{C_{qm}}{\left\{ \frac{T_{t1}}{T_{t0}} - \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} \right\}} = \frac{\frac{\gamma+1}{\gamma-1} (M_s^*)^{-1}}{\left\{ (\gamma+1) \left(1 + \frac{T_{t1}}{T_{t0}} \right) \right\}^{1/2}} \quad (A-12)$$

Consider the case where P_{t1}/P_{t0} , T_{t1}/T_{t0} , and P_{t5}/P_{t0} are fixed. This equation shows that under these conditions C_{pm} and C_{qm} will be a minimum when the mass flow ratio, $\dot{m}_1/\dot{m}_3 = \chi$, is a maximum. Also, it is noted that this equation is invariant for both the constant pressure and constant area cases, although the maximum value of χ may differ in the two cases.

d. Efficiency

Another performance parameter of some interest is the mechanical efficiency which is, of course, based on energy consideration, namely,

$$\eta_M = \frac{\text{Final available energy (Isentropic)}}{\text{Initial available energy (Isentropic)}}$$

$$\eta_M = \frac{\dot{m}_5 (H_5 - h_5)}{\dot{m}_3 \Delta H_c} = \frac{\rho_5 A_5 V_5}{\rho_3 A_3 V_3} \cdot \frac{V_5^2/2gJ}{c_p \Delta T_c}$$

$$\eta_M = \frac{1}{\frac{\rho_3 A_3 V_3}{\rho_5 A_5 V_5}} \cdot \frac{\frac{V_5}{a_5^*} \frac{c_p \Delta T_c}{V_5} \cdot \frac{V_5^2/2gJ}{a_5^*}}{\frac{V_5}{a_5^*}} = \frac{V_5}{C_{pm} a_5^*}$$

$$\eta_M = \frac{a_5^* V_5}{C_{pm} a_5^* a_5^*} = \frac{1}{C_{pm}} \sqrt{\frac{T_{t5}}{T_{t0}}} M_s^*$$

Substituting for C_{PM} from (A-8)

$$\eta_M = \frac{(\chi+1) \sqrt{\frac{T_{t5}}{T_{t0}}} M_s^*}{\left\{ \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}^{\frac{\gamma+1}{\gamma-1}}} \sqrt{\frac{T_{t5}}{T_{t0}}} M_s^*$$

Substituting for T_{t5}/T_{t0} from the energy equation (A-31).

$$\eta_M = \frac{\left(\frac{\gamma-1}{\gamma+1} \right) M_s^{*2}}{\left\{ \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}} \left(\chi + \frac{T_{t1}}{T_{t0}} \right) \quad (A-13)$$

Since, $p_s = P_{t0}$, then

$$\frac{\gamma-1}{\gamma+1} M_s^{*2} = 1 - \left(\frac{P_{t0}}{P_{t1}} \right)^{\frac{\gamma-1}{\gamma}}$$

Substituting for $\frac{\gamma-1}{\gamma+1} M_s^{*2}$ into (A-13)

Therefore:

$$\eta_M = \frac{\left\{ 1 - \left(\frac{P_{t0}}{P_{t1}} \right)^{\frac{\gamma-1}{\gamma}} \right\}}{\left\{ \left(\frac{P_{t1}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}} \left(\chi + \frac{T_{t1}}{T_{t0}} \right) \quad (A-14)$$

This equation shows that η_M , like the other performance parameters, will be optimum when χ is a maximum for specified conditions of pressure and temperature.

3. Development of constant pressure jet pump theory

An explicit solution to the equations of continuity, energy, and momentum is possible when the following parameters are considered fixed:

- P_{t5}/P_{t0} the ratio of the demanded discharge total pressure to the ambient pressure.
- P_{t1}/P_{t0} the ratio of the primary (jet) total pressure to the ambient pressure.
- T_{t1}/T_{t0} the ratio of the primary (jet) total temperature to the ambient temperature.
- M_s^* the dimensionless secondary velocity.

The usual isentropic relationships are used to solve for conditions in the primary (jet) system and in the secondary jet system up to the mixing tube entrance.

The critical value of M_2^* for choking of the primary nozzle can be determined from the isentropic relationships

$$\frac{P_1}{P_u} = \frac{P_2}{P_{t2}} = \left[1 - \frac{\gamma-1}{\gamma+1} M_2^{*2} \right]^{\frac{\gamma}{\gamma-1}} \quad (\text{A-15})$$

$$\frac{P_2}{P_{t2}} = \frac{P_2}{P_{t2}} = \left[1 - \frac{\gamma-1}{\gamma+1} M_2^{*2} \right]^{\frac{\gamma}{\gamma-1}} \quad (\text{A-16})$$

For sub-critical flow, $M_2^* < 1.0$ and $M_2^* < M_{2cr}^*$, the primary flow is unchoked and $P_1 = P_2$. M_{2cr}^* is the critical secondary velocity ratio at which M_2^* becomes sonic for supply pressure ratios less 1.89. Multiplying P_2/P_{t2} by P_u/P_2 from (A-15) and (A-16) results in

$$\frac{P_u}{P_{t2}} = \left\{ \frac{1 - \frac{\gamma-1}{\gamma+1} M_{2cr}^{*2}}{1 - \frac{\gamma-1}{\gamma+1} M_2^{*2}} \right\}^{\frac{\gamma}{\gamma-1}} \quad (\text{A-17})$$

Solving for M_2^* from (A-17)

$$M_2^* = \left\{ \frac{\gamma+1}{\gamma-1} \left[1 - \frac{\left[1 - \frac{\gamma-1}{\gamma+1} M_{2cr}^{*2} \right]^{\frac{\gamma}{\gamma-1}}}{\left(\frac{P_u}{P_{t2}} \right)^{\frac{\gamma}{\gamma-1}}} \right] \right\}^{\frac{1}{2}} \quad (\text{A-18})$$

Also

$$\frac{P_1}{P_{t2}} = \frac{P_2}{P_{t2}} = \frac{P_2}{P_{t2}} = \left[1 - \frac{\gamma-1}{\gamma+1} M_2^{*2} \right]^{\frac{\gamma}{\gamma-1}}$$

At critical conditions, $M_2^* = 1$, $M_2^* = M_{2cr}^*$, and $P_1 = P_2$. It is possible to determine the critical M_2^* as a function of P_u/P_{t2} from (A-17)

$$\frac{2}{\gamma+1} \left(\frac{P_1}{P_0} \right)^{\frac{\gamma-1}{\gamma}} = 1 - \frac{\gamma-1}{\gamma+1} M_{2cr}^{*2}$$

$$M_{2cr}^* = \left\{ \frac{\gamma+1}{\gamma-1} \left[1 - \frac{2}{\gamma+1} \left(\frac{P_1}{P_0} \right)^{\frac{\gamma-1}{\gamma}} \right] \right\}^{\frac{1}{2}} \quad (A-19)$$

For super-critical flow, $M_3^* = 1$, $M_2^* > M_{2cr}^*$, and $p_2 \neq p_3$. For supply pressure ratios, P_1/P_0 , greater than 1.89, $M_3^* = 1.0$ for all conditions. This renders the concept of M_{2cr}^* meaningless and p_2 is not equal to p_3 , specifically $p_2 < p_3$.

$$\frac{P_1}{P_0} = \left(\frac{P_2}{P_1} \right) \left(\frac{P_1}{P_0} \right)$$

$$\frac{P_1}{P_0} = \frac{P_2}{P_0} = \left[1 - \frac{\gamma-1}{\gamma+1} M_3^{*2} \right]$$

Since $M_3^* = 1.0$,

$$\frac{P_2}{P_0} = \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}} \left(\frac{P_1}{P_0} \right) \quad (A-20)$$

and

$$\frac{P_2}{P_0} = \left[1 - \frac{\gamma-1}{\gamma+1} M_2^{*2} \right]^{\frac{\gamma}{\gamma-1}}$$

The basic condition for constant pressure mixing is:

$$p_2 = p_4 \quad (A-21)$$

Therefore $\frac{P_2}{P_0} = \left(\frac{P_4}{P_0} \right) \left(\frac{P_0}{P_4} \right)$ since $P_{t4} = P_{t0}$

Substituting the isentropic relationships for p_2/P_0 and P_{t5}/P_{t0}

($P_3 = P_{t0}$) and raising to the $\frac{\gamma-1}{\gamma}$ power.

$$\left[1 - \frac{\gamma-1}{\gamma+1} M_2^{*2} \right] = \left(\frac{P_{t5}}{P_{t0}} \right)^{\frac{\gamma-1}{\gamma}} \left[1 - \frac{\gamma-1}{\gamma+1} M_4^{*2} \right]$$

Hence M_4^* can be determined from

$$M_4^* = \left\{ \frac{\gamma+1}{\gamma-1} \left[1 - \frac{\left[1 - \frac{\gamma-1}{\gamma+1} V_1^2 \right]}{\left(\frac{P_3}{P_0} \right)^{\frac{\gamma-1}{\gamma}}} \right] \right\}^{\frac{1}{2}} \quad (A-22)$$

Analyzing the Equations (A-18) and (A-22), it is seen that M_3^* and M_4^* are fixed when P_1/P_0 , P_3/P_0 and M_2^* are fixed. T_u/T_0 does not appear in either equation.

The three basic equations that have to be satisfied for constant pressure mixing ($P_2 = P_4$) are continuity, energy and momentum.

Continuity:

$$\dot{m}_2 + \dot{m}_3 = \dot{m}_4 \quad (A-23)$$

Energy:

$$\dot{m}_2 H_2 + \dot{m}_3 H_3 = \dot{m}_4 H_4 \quad (A-24)$$

Momentum:

$$(\dot{m}_3 + \dot{m}_2) V_4 - (\dot{m}_3 V_3 + \dot{m}_2 V_2) = (p_3 - p_2) A_3 \quad (A-25)$$

It is noted that the resultant pressure forces in (A-25) are zero in the constant pressure case when $M_3^* = 1$ and $M_2^* = M_{2cr}^*$ since $p_2 = p_3$. When $M_3^* = 1$ and $M_2^* > M_{2cr}^*$ then $p_2 < p_3$. The continuity equation (A-23) can be rearranged as follows:

$$\frac{\dot{m}_2}{\dot{m}_3} + 1 = \frac{\dot{m}_4}{\dot{m}_3}$$

Let $x = \dot{m}_2/\dot{m}_3$

So,

$$\begin{aligned} x+1 &= \frac{P_3}{P_0} \frac{A_4}{A_3} \frac{M_3^*}{M_1^*} \frac{a_1^*}{a_3^*} \\ \frac{A_4}{A_3} &= (x+1) \frac{P_3}{P_0} \frac{M_1^*}{M_3^*} \frac{a_1^*}{a_3^*} \end{aligned} \quad (A-26)$$

and from the isentropic relationship:

$$\frac{P}{P_t} = \left[1 - \frac{\gamma-1}{\gamma+1} M^2 \right]^{\frac{\gamma}{\gamma-1}}$$

$$\frac{P_3}{P_4} = \frac{P_{t3}}{P_{t4}} \left\{ \frac{1 - \frac{\gamma-1}{\gamma+1} M_3^2}{1 - \frac{\gamma-1}{\gamma+1} M_4^2} \right\}^{\frac{\gamma}{\gamma-1}} = \frac{P_{t3}}{P_{t4}} \left\{ \frac{1 - \frac{\gamma-1}{\gamma+1} M_3^2}{1 - \frac{\gamma-1}{\gamma+1} M_4^2} \right\}^{\frac{\gamma}{\gamma-1}}$$

$$\frac{P_{t3}}{P_{t4}} = \frac{P_{t3}}{P_{t5}} \frac{T_{t5}}{T_{t4}} = \frac{P_{t3}}{P_{t0}} \frac{T_{t5}}{T_{t0}}$$

Hence,

$$\frac{P_3}{P_4} = \frac{T_{t5}}{T_{t0}} \frac{P_{t3}}{P_{t0}} \frac{1}{\frac{P_{t4}}{P_{t0}}} \left\{ \frac{1 - \frac{\gamma-1}{\gamma+1} M_3^2}{1 - \frac{\gamma-1}{\gamma+1} M_4^2} \right\}^{\frac{\gamma}{\gamma-1}}$$

$$\frac{a_3^*}{a_4^*} = \left(\frac{T_{t3}}{T_{t4}} \right)^{\frac{1}{2}} = \left(\frac{T_{t3}}{T_{t4}} \frac{T_{t4}}{T_{t0}} \frac{T_{t0}}{T_{t4}} \right)^{\frac{1}{2}} = \left(\frac{T_{t3}}{T_{t0}} \frac{T_{t0}}{T_{t4}} \right)^{\frac{1}{2}}$$

Substituting for P_3/P_4 and a_3^*/a_4^* into Equation (A-26).

$$\frac{A_4}{A_3} = (\gamma+1) \left(\frac{T_{t3}}{T_{t0}} \right)^{\frac{1}{2}} \frac{P_{t3}}{P_{t0}} \frac{1}{\left(\frac{P_{t4}}{P_{t0}} \right)^{\frac{1}{2}}} \left\{ \frac{1 - \frac{\gamma-1}{\gamma+1} M_3^2}{1 - \frac{\gamma-1}{\gamma+1} M_4^2} \right\}^{\frac{\gamma}{\gamma-1}} \frac{M_3^*}{M_4^*}$$

It will be shown from the energy equation (A-31) that;

$$\frac{T_{t5}}{T_{t0}} = \frac{T_{t4}}{T_{t0}} = \frac{\left(\frac{T_{t3}}{T_{t0}} + \gamma \right)}{1 + \gamma}$$

Substituting for T_{t4}/T_{t0} , then A_4/A_3 becomes;

$$\frac{A_4}{A_3} = \left[(\gamma+1) \left(\frac{T_{t3}}{T_{t0}} + \gamma \right) \right]^{\frac{1}{2}} \frac{P_{t3}}{P_{t0}} \frac{1}{\left(\frac{P_{t4}}{P_{t0}} \right)^{\frac{1}{2}}} \left\{ \frac{1 - \frac{\gamma-1}{\gamma+1} M_3^2}{1 - \frac{\gamma-1}{\gamma+1} M_4^2} \right\}^{\frac{\gamma}{\gamma-1}} \frac{M_3^*}{M_4^*} \quad (A-27)$$

This is the dimensionless form of the continuity equation.

Similarly replacing M_4^* by M_5^* and noting that $T_{e5} = T_{e4}$ and $P_{e5} = P_{e4}$ then;

$$\frac{A_5}{A_3} = \left[(\gamma + 1) \left(\frac{T_{t1}}{T_{t0}} + \gamma \right) \right]^{\frac{1}{2}} \frac{\frac{P_{t4}}{P_{t0}}}{\frac{P_{t5}}{P_{t0}} \left(\frac{T_{t1}}{T_{t0}} \right)^{\frac{1}{2}}} \left\{ \frac{1 - \frac{\gamma-1}{\gamma+1} M_3^{*2}}{1 - \frac{\gamma-1}{\gamma+1} M_5^{*2}} \right\}^{\frac{1}{\gamma-1}} \frac{M_3^*}{M_5^*} \quad (A-28)$$

Similarly for A_6/A_3 replacing M_5^* by $M_6^* = 1$

$$\frac{A_6}{A_3} = \left[(\gamma + 1) \left(\frac{T_{t1}}{T_{t0}} + \gamma \right) \right]^{\frac{1}{2}} \frac{\frac{P_{t4}}{P_{t0}}}{\frac{P_{t6}}{P_{t0}} \left(\frac{T_{t1}}{T_{t0}} \right)^{\frac{1}{2}}} \left\{ \frac{1 - \frac{\gamma-1}{\gamma+1} M_3^{*2}}{\frac{2}{\gamma+1}} \right\}^{\frac{1}{\gamma-1}} M_3^* \quad (A-29)$$

From the mass flow ratio γ , it is possible to determine A_2/A_3 .

$$\frac{\dot{m}_2}{\dot{m}_3} = \gamma = \frac{\rho_2 A_2 a_2^* M_2^*}{\rho_3 A_3 a_3^* M_3^*}$$

and:

$$\frac{A_2}{A_3} = \gamma \frac{\frac{P_{t4}}{P_{t0}}}{\left(\frac{T_{t1}}{T_{t0}} \right)^{\frac{1}{2}}} \left\{ \frac{1 - \frac{\gamma-1}{\gamma+1} M_3^{*2}}{1 - \frac{\gamma-1}{\gamma+1} M_2^{*2}} \right\}^{\frac{1}{\gamma-1}} \frac{M_3^*}{M_2^*} \quad (A-30)$$

Consider the energy equation, (A-24).

$$\dot{m}_2 c_p T_{t2} + \dot{m}_3 c_p T_{t3} = \dot{m}_4 c_p T_{t4}$$

$$\dot{m}_2 T_{t2} + \dot{m}_3 T_{t3} = \dot{m}_4 T_{t4}$$

$$\frac{\dot{m}_2}{\dot{m}_3} + \frac{T_{t1}}{T_{t0}} = \left(\frac{\dot{m}_4}{\dot{m}_3} + 1 \right) \frac{T_{t4}}{T_{t0}}$$

The non-dimensional form of energy equation is

$$\left(\frac{T_{t1}}{T_{t0}} + \gamma \right) = (1 + \gamma) \frac{T_{t4}}{T_{t0}} \quad (A-31)$$

Now consider the momentum equation (A-25). Substituting for the velocities and rearranging

$$\dot{m}_3 a_3^* M_3^* + \dot{m}_2 a_2^* M_2^* + (p_3 - p_2) A_3 = (\dot{m}_3 + \dot{m}_2) a_4^* M_4^*$$

Dividing both sides by $\dot{m}_3 a_3^*$.

$$\frac{a_4^*}{a_3^*} M_3^* + \left(\frac{\dot{m}_2}{\dot{m}_3}\right) \frac{a_2^*}{a_3^*} M_2^* + \frac{(p_3 - p_2) A_3}{a_3^* \dot{m}_3} = \left(1 + \frac{\dot{m}_2}{\dot{m}_3}\right) \frac{a_4^*}{a_3^*} M_4^* \quad (A-32)$$

Let

$$D = \frac{(p_3 - p_2) A_3}{a_3^* \dot{m}_3}$$

D can be expressed as follows

$$D = \left(1 - \frac{p_2}{p_3}\right) \frac{p_3}{a_3^*} \frac{1}{\rho_3 V_3} = \left(1 - \frac{p_2}{p_3}\right) \frac{p_3}{a_3^*} \frac{1}{\rho_3 a_3^* M_3^*}$$

$$D = \left(1 - \frac{p_2}{p_3}\right) \frac{1}{a_3^*} \frac{1}{M_3^*} \frac{1}{\sqrt{T_{t_3}}}$$

$$D = \left(1 - \frac{p_2}{p_3}\right) \frac{\gamma+1}{2\gamma} \sqrt{\frac{T_{t_1}}{T_{t_3}}} \frac{1}{M_3^*} \left[1 - \frac{\gamma-1}{\gamma+1} M_3^{*2}\right]$$

For $M_3^* \leq 1$ and $M_2^* \leq M_{2cr}$, $p_2 = p_3$; therefore $D = 0$.

For $M_3^* = 1$ and $M_2^* > M_{2cr}$, $p_2 \neq p_3$.

Therefore:

$$D = \left(1 - \frac{p_2}{p_3}\right) \frac{\gamma+1}{2\gamma} \sqrt{\frac{T_{t_1}}{T_{t_3}}} \left[\frac{2}{\gamma+1}\right] \neq 0$$

$$D = \frac{1}{\gamma} \sqrt{\frac{T_{t_1}}{T_{t_3}}} \left(1 - \frac{p_2}{p_3}\right)$$

$$D = \frac{1}{\gamma} \sqrt{\frac{T_{t_1}}{T_{t_3}}} \left(1 - \frac{p_2}{p_3}\right)$$

$$D = \frac{1}{\gamma} \sqrt{\frac{T_{t1}}{T_{t0}}} \left(1 - \frac{\frac{P_t}{P_{t0}}}{\left(\frac{P_0}{P_{t1}}\right)\left(\frac{P_{t1}}{P_{t0}}\right)} \right)$$

$$D = \frac{1}{\gamma} \sqrt{\frac{T_{t1}}{T_{t0}}} \left\{ 1 - \left(\frac{1 - \frac{\gamma-1}{\gamma+1} M_1^{*2}}{\frac{2}{\gamma+1}} \right)^{\frac{\gamma}{\gamma-1}} \frac{P_{t0}}{P_{t1}} \right\}$$

Let

$$\xi = \frac{1}{\gamma} \left\{ 1 - \left(\frac{1 - \frac{\gamma-1}{\gamma+1} M_1^{*2}}{\frac{2}{\gamma+1}} \right)^{\frac{\gamma}{\gamma-1}} \frac{P_{t0}}{P_{t1}} \right\}$$

$$D = \sqrt{\frac{T_{t1}}{T_{t0}}} \xi$$

Substituting into Equation (A-32), the dimensionless momentum equation is

$$\sqrt{\frac{T_{t1}}{T_{t0}}} M_3^* + \kappa M_1^* + \sqrt{\frac{T_{t1}}{T_{t0}}} \xi = (1+\kappa) \sqrt{\frac{T_{t1}}{T_{t0}}} M_4^* \quad (\text{A-33})$$

By fixing P_5/P_0 , P_1/P_0 , T_{t1}/T_{t0} , and M_1^* , it is possible to obtain an explicit solution from the continuity, energy, and momentum equations since there remains only three unknowns κ , T_{t4}/T_{t0} , and A_4/A_3 .

The energy and momentum equations are solved simultaneously for κ and T_{t4}/T_{t0} . By eliminating T_{t4}/T_{t0} between the two equations, the following quadratic is obtained.

$$(M_4^{*2} - M_1^{*2})\gamma^2 + \left\{ \left(\frac{T_{t1}}{T_{t0}} + 1 \right) M_4^{*2} - 2 \sqrt{\frac{T_{t1}}{T_{t0}}} M_1^* (M_3^* + \xi) \right\} \kappa + \frac{T_{t1}}{T_{t0}} \left\{ M_4^{*2} - (M_3^* + \xi)^2 \right\} = 0$$

This equation can be solved by using the quadratic formula

$$\gamma = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

where

$$a = M_4^{*2} - M_2^{*2}$$

$$b = \left(\frac{T_{41}}{T_{40}} + 1 \right) M_4^{*2} - 2 \sqrt{\frac{T_{41}}{T_{40}}} M_2^{*2} (M_3^{*2} + \xi)$$

$$c = \frac{T_{41}}{T_{40}} \left\{ M_4^{*2} - (M_3^{*2} + \xi) \right\}$$

Once γ is determined, then T_{41}/T_{40} can be determined from the energy equation and A_4/A_3 can be determined from the continuity equation. In addition, it is now possible to determine the area ratios A_2/A_3 , A_4/A_3 , and A_4/A_2 .

4. Development of constant area theory:

This development was provided by Dr. Gawain in Reference 2. Basically, it is a process to non-dimensionalize the equations of continuity, energy and momentum. These equations are initially in the following form:

Continuity:

$$\dot{m}_2 + \dot{m}_3 = \dot{m}_4 \quad (A-34)$$

Energy:

$$\dot{m}_2 H_2 + \dot{m}_3 H_3 = \dot{m}_4 H_4 \quad (A-35)$$

Momentum:

$$\dot{m}_4 V_4 - \dot{m}_2 V_2 - \dot{m}_3 V_3 = p_2 A_2 + p_3 A_3 - p_4 A_4 \quad (A-36)$$

First, consider the continuity equation (A-34). Expanding this equation gives the relationship:

$$\rho_1 A_1 M_1^* a_1^* + \rho_2 A_2 M_2^* a_2^* = \rho_3 A_3 M_3^* a_3^*$$

It must be understood that $A_1 + A_2 = A_3$ for the constant area case.

The general relationship for the ratio of mass flow rate to the area is:

$$\dot{m}/A = \rho V = \rho a^* M^* = \frac{\rho}{\rho_t} \rho_t a^* M^* \quad (A-37)$$

Using the isentropic relationship for the density ratio and expanding ρa^* yields:

$$\dot{m}/A = \left\{ 1 - \frac{\gamma-1}{\gamma+1} M^{*2} \right\}^{\frac{1}{\gamma-1}} \frac{\rho_t}{\gamma R T_t} \left\{ \frac{2\gamma}{\gamma+1} \gamma R T_t \right\}^{\frac{1}{2}} M^*$$

Let

$$K_1 = \left\{ \frac{2\gamma}{(\gamma+1)\gamma R} \right\}^{\frac{1}{2}}$$

The final form of Equation (A-37) is:

$$\dot{m}/A = K_1 \rho_t (T_t)^{-\frac{1}{2}} \left(1 - \frac{\gamma-1}{\gamma+1} M^{*2} \right)^{\frac{1}{\gamma-1}} M^*$$

It is convenient to define a velocity function $\phi(M^*)$:

$$\phi(M^*) \equiv \frac{\rho}{\rho_t} M^* = \left(1 - \frac{\gamma-1}{\gamma+1} M^{*2} \right)^{\frac{1}{\gamma-1}} M^*$$

Solving for the mass flow rate gives:

$$\dot{m} = K_1 A \rho_t (T_t)^{-\frac{1}{2}} \phi(M^*) \quad (A-38)$$

Now substituting Equation (A-38) into Equation (A-34).

$$K_1 A_1 \rho_{t1} (T_{t1})^{-\frac{1}{2}} \phi(M_1^*) + K_1 A_2 \rho_{t2} (T_{t2})^{-\frac{1}{2}} \phi(M_2^*) = K_1 A_3 \rho_{t3} (T_{t3})^{-\frac{1}{2}} \phi(M_3^*)$$

It can be seen from Figure 2 that the total state conditions at stations 1 and 3, and stations 0 and 2 are equal. The preceding equation is non-dimensionalized by dividing through by $K_1 A_3 \rho_{t0} (T_{t0})^{-\frac{1}{2}}$. This results in the final non-dimensional form of the continuity equation.

$$y \phi(M_2^*) + \frac{P_{t1}}{P_{t0}} \sqrt{\frac{T_{t0}}{T_{t1}}} \phi(M_3^*) = (1+y) \frac{P_{t4}}{P_{t0}} \sqrt{\frac{T_{t0}}{T_{t4}}} \phi(M_4^*) \quad (A-39)$$

The energy equation (A-35) for the constant area jet pump can be expanded to yield the relationship:

$$\rho_2 A_2 V_2 c_p T_{t2} + \rho_3 A_3 V_3 c_p T_{t3} = \rho_4 A_4 V_4 c_p T_{t4} \quad (A-40)$$

In general, the term $\rho V T_t$ can be expanded into the form:

$$\rho V T_t = \rho \frac{P}{\rho} a^* M^* T_t = T_t \rho_t a^* \phi(M^*)$$

and,

$$T_t \rho_t a^* = \left\{ \frac{2r}{(r+1) \kappa g} \right\}^{\frac{1}{2}} P_t (T_t)^{\frac{1}{2}} = K_1 P_t (T_t)^{\frac{1}{2}}$$

Therefore;

$$\rho V T_t = K_1 P_t (T_t)^{\frac{1}{2}} \phi(M^*)$$

Substituting this relationship into Equation (A-40) and non-dimensionalizing in the same manner as before results in the final form of the energy equation.

$$y \phi(M_2^*) + \frac{P_{t1}}{P_{t0}} \sqrt{\frac{T_{t0}}{T_{t1}}} \phi(M_3^*) = (1+y) \frac{P_{t4}}{P_{t0}} \sqrt{\frac{T_{t0}}{T_{t4}}} \phi(M_4^*) \quad (A-41)$$

The specific heat at constant pressure (c_p) is assumed to be constant.

Finally, considering the momentum equation (assuming steady, frictionless flow and the velocities are uniform at stations 2, 3, and 4):

$$\dot{m}_4 V_4 = \dot{m}_2 V_2 - \dot{m}_3 V_3 = p_2 A_2 + p_3 A_3 - p_4 A_4 \quad (A-36)$$

Rearranging and substituting for \dot{m} and A_4 gives;

$$A_2(p_2 + \rho_2 V_2^2) + A_3(p_3 + \rho_3 V_3^2) = (A_2 + A_3)(p_4 + \rho_4 V_4^2)$$

Now consider $(p + \rho V^2)$ in general. This factor can be expanded to yield,

$$p + \rho V^2 = p + \frac{p}{R_g T} a^2 M^2 = p \left(1 + \frac{\gamma}{\gamma+1} \frac{T_c}{T} M^2 \right)$$

where

$$\frac{T_c}{T} = \left(1 + \frac{\gamma-1}{\gamma+1} M^2 \right)^{-1}$$

and

$$\frac{p}{R} = \left(1 + \frac{\gamma-1}{\gamma+1} M^2 \right)^{\frac{\gamma}{\gamma-1}}$$

Therefore;

$$p + \rho V^2 = \frac{p}{R} R \left\{ \frac{1 + M^2}{1 + \frac{\gamma-1}{\gamma+1} M^2} \right\} = R \left\{ 1 + \frac{\gamma-1}{\gamma+1} M^2 \right\}^{\frac{1}{\gamma-1}} \left\{ 1 + M^2 \right\}$$

It is helpful to define $\psi(M^2)$ as;

$$\psi(M^2) = \left\{ 1 + \frac{\gamma-1}{\gamma+1} M^2 \right\}^{\frac{1}{\gamma-1}} \left\{ 1 + M^2 \right\}$$

So the final form of the term $(p + \rho V^2)$ is:

$$p + \rho V^2 = R \psi(M^2) \quad (\text{A-42})$$

Substituting Equation (A-42) into Equation (A-36) and non-dimensionalizing yields the momentum equation:

$$\gamma \psi(M_1^2) + \frac{P_{t1}}{P_{t0}} \psi(M_3^2) = (1 + \gamma) \frac{P_{t4}}{P_{t0}} \psi(M_4^2) \quad (\text{A-43})$$

From the continuity equation, the mass flow ratio is determined to be;

$$\dot{m}_2 / \dot{m}_3 = \chi = \gamma \frac{P_{t0}}{P_{t1}} \sqrt{\frac{T_{t1}}{T_{t0}}} \frac{\phi(M_2^2)}{\phi(M_3^2)} \quad (\text{A-44})$$

And from the energy equation, the mixing tube discharge temperature ratio is:

$$\frac{T_{t4}}{T_{t0}} = \frac{\gamma + T_{t1}/T_{t0}}{\gamma + 1} \quad (\text{A-43})$$

Dividing Equation (A-41) by Equation (A-43) results in a quadratic relationship in terms of M_4^* .

$$\frac{\phi(M_4^*)}{\psi(M_4^*)} = \sqrt{\frac{T_{t0}}{T_{t4}}} \left\{ \frac{y \phi(M_2^*) + \frac{P_{t1}}{P_{t0}} \sqrt{\frac{T_{t1}}{T_{t0}}} \phi(M_3^*)}{y \psi(M_2^*) + \frac{P_{t1}}{P_{t0}}} \psi(M_3^*) \right\} \quad (\text{A-46})$$

where,

$$\frac{\phi(M_4^*)}{\psi(M_4^*)} = \frac{M_4^*}{1 + M_4^{*2}}$$

It can be shown that the right side of Equation (A-46) must be equal to or less than one-half. Let,

$$f(M_4^*) = \frac{M_4^*}{1 + M_4^{*2}} \quad (\text{A-47})$$

Taking the derivative of this function with respect to M_4^* .

$$\frac{df(M_4^*)}{dM_4^*} = \frac{1 - M_4^{*2}}{(1 + M_4^{*2})^2}$$

Setting the derivative equal to zero and solving for M_4^* yields,

$$M_4^{*2} = 1$$

Choosing the positive root and substituting into Equation (A-47) gives the maximum value of the function $f(M_4^*)$.

$$f(M_4^*)_{\max} = 1/2$$

Therefore;

$$f(M_4^*) \leq 1/2$$

This condition is a limit for the constant area case.

Solutions for optimum constant area jet pumps are contained in Appendix C. An iterative procedure was used to satisfy the boundary condition that P_{t4}/P_{t0} must equal P_{t5}/P_{t0} . This was accomplished by varying the area ratio A_2/A_3 until P_{t4}/P_{t0} was equal to the specified value of P_{t5}/P_{t0} . It is emphasized that each line of the computer results represents the optimum pump of a family of constant area jet pumps.

APPENDIX B

Itemized cost estimate

The following pages show the itemized cost estimate for various test facility configurations. The numbers for each item up to item 22 corresponds to the numbers shown on Figures 26 through 34. The labor estimate was made by Mr. Robert Besel, Laboratory Supervisor for the Aeronautical Engineering Department.

Material estimates were solicited from numerous companies. The code used for denoting the company is as follows:

Code	Company
MC	McMaster-Carr Supply Co.
DU	Ducommun Metals and Supply Co.
RE	Reliance Steel Co.
ME	Meriam Instrument Co.
PA	Pacific Metals Co.
RY	Ryerson Steel
GC	General Controls Corp.
USC	United Sensor and Control Corp.

The total cost estimates are included in the following itemized cost estimates.

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ITEM	Units	Unit Price	Maximum Price	Minimum Price
1. Pressure tank: Stock No. - 437P611(MC)	1		149.60	149.60
2. Flange: ARA 150 Forged Steel Slip-on flange, NPS-4 in. (DU)	3	20.00	60.00	60.00
3. Pipe: ASTM A-53 SCH 40 4 inch pipe (HE)	9 ft 15 ft	1.80/ft	18.20	36.20
4. Orifice plates and flanges: (MA) 4 in. pipe-Series 30 flange orifice plate 6 in. pipe-Series 30 flange orifice plate	1 1 1 1		32.00 10.00 50.50 11.25	32.00 10.00 50.50 11.25
5. Elbow: 90° radius elbow for 4 in. pipe, No. 406 (DU)	1		19.50	19.50
6. Pipe: Type 304 SCH 40 Stainless steel pipe (PA) 6 inch pipe 5 inch pipe 4 inch pipe	8 ft 10 ft 2 ft 8 ft	39.44/ft 35.53/ft 35.53/ft 26.22/ft	316.60 355.98 210.27	71.06
7. Flange: Type 316 MSS Stainless steel slip-on flanges (DU) NPS-4 inch NPS-5 inch NPS-6 inch	6 6 2 6	21.37 33.69 33.69 36.97	128.22 268.92 221.82	67.38

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ITEM	Units	Unit Price	Maximum Price	Minimum Price
8. Nozzle: Type 304 Stainless steel round bar, 6 in. dia. Spec. QQ-S-763B (PA)	144 lb 46 lb	.87/lb .87/lb	128.58	42.86
9. Annulus - Hot rolled steel ring, O.D. 10" (NY) I.D. 4 in-3 in. thick I.D. 5 in-2 1/4 in. thick I.D. 6 in-2 in. thick	1 1 1		63.08 45.06 36.05	63.08
10. Backing Ring - ASA 150 lb forged steel blind flange NPS - 10 inches (DU)	1		58.00	
11. Sheet: cold rolled 12 gage steel (48" x 120") (PA)	175 lb	.22/lb	39.20	39.20
12. Flange: ASA 150 forged steel slip-on flange, (DU) NPS - 6 in. NPS - 4 in.	2 4	27.00 20.00	54.00	54.00 80.00
13. Pipe: ASTM-A-53, SCH 40, 6 inch steel pipe (RE)	12 ft	2.85/ft	36.20	36.20
14. Flange: Type 316 MSS stainless steel slip-on "special order" O.D. 16". NPS 4,5,6 (DU)	3 1	90.38	271.14	90.38
15. Flange: ASA 150 forged steel slip-on reducing flange (DU) 4 x 11" O.D. 5 x 11" O.D. 6 x 11" O.D.	1 1 2	49.00 49.00 98.00		

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ITEM	Units	Unit Price	Maximum Price	Minimum Price
16. Tubing: AISI MT-1015, 10 Gage dia. 2.375", length 24' (RE)	77 lb	.67/lb	52.59	52.59
17. Turn Buckles: Drop forged Type 302 (DU)	6	1.10	6.60	6.60
18. Channel: Aluminum 6063-T5 Die #2388, Length 48' (PA)	20 lb	1.66/lb	32.20	
19. Bar: Aluminum 2024-T4 rectangular bar, 1/2" x 2" x 36" (PA)	21.5 lb	2.08/lb	44.72	
20. Casters: SB-4689 BASSICK (DU)	8	6.20	49.60	
21. Mounting plates: Mixing tube actuator mounts (local)	48 8	1.00	48.00	8.00
22. Valve: Electrically actuated remote controlled butterfly valve (CC)	1		326.10	326.10
23. Tubing: Copper tubing, O.D. 1/2" 50' coil (DU)	4	6.06	24.24	
24. Tubing: Imperial poly-flo tubing, Stock #66-P (DU)	200 ft	.09/ft	17.80	
25. Wall Tape: static pressure mixing tube wall taps (DU)	72	.50	36.00	

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ITEM	Units	Unit Price	Maximum Price	Minimum Price
26. Connectors: wall tap to copper tube connectors, No. 768 - FS (DU)	72	1.89	136.08	
27. Connectors: copper tube to plastic tube connectors, No. 262-PC (DU)	72	0.26	18.73	
28. Plugs: Dryseal threaded steel pressure plugs (DU)	50	.15	7.60	
29. Probe: Kiel - temperature probe, Order # KT-14-C-ca-3 (USC)	3	30.00	90.00	90.00
30. Receiver: USC - 1396-6	2	380.00	760.00	
31. Mechanical Cables: USC-1103-30	2	68.00	136.00	
32. Collets: USC-1499-125	2	1.50	3.00	3.00
Proposal For Remote Electrically Controlled System				
33. Transmitter: USC-125I	1		310.00	
34. Motor Box: USC-1462-B	1		675.00	
Total Material Cost For Remote Electrical Control System			5555.42	

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ITEM	Units	Unit Price	Maximum Price	Minimum Price
Alternate Proposal For Remote Manual Controlled System				
35. Transmitter: USC-1793 replaces items 33 and 34	1		220.00	
Total Material Cost For Remote Manual Control			4787.42	
Instrumentation Necessary For Minimum Cost System				
36. Manual traverse unit: replaces items 30, 31, 33, 34, and 35 USC-1108-12	2	72.00		144.00
Total Cost For Minimum Priced System				1601.50

Labor Estimate

	Man Hours
1. Drill and tap flanges for primary system	7
2. Install mass flow orifice plates and flanges	7
3. Fabrication of nozzles	90
4. Fabrication of mixing tube sections	108
5. Fabrication of bell-mouth annulus	49
6. Fabrication of secondary air plenum	32
7. Drill and tap flanges for secondary system	2
8. Install butterfly valve	4
9. Fabricate mixing tube stanchions	6
10. Fabricate handling cart	16
11. Assemble complete system and hook-up instrumentation	42
Total estimated man hours	362

APPENDIX C

Computer Results

This appendix contains computer solutions for a set of optimum constant area and constant pressure heated jet pumps for the conditions specified. Each line of data represents the parameters defining one optimum pump. The momentum coefficient is constant for each page of data.

Also included is a sample computer program and solutions for one jet pump with a fixed area ratio, $A_2/A_3 = 2.16$, and fixed supply conditions. Each line represents the theoretical performance for the conditions specified.

The symbols used in the computer programs are as follows:

- PT5P0 = the ratio of the demanded discharge total pressure to the ambient pressure.
- PnPn = pressure ratio, station (n) to (n).
- TnTn = temperature ratio, station (n) to (n).
- AnAn = area ratio, station (n) to (n).
- SACEn = velocity ratio (M^*) at station (n).
- TT5T0 = ratio of the discharge total temperature to the ambient temperature.
- WWSJ = mass flow ratio, secondary to primary.
- G = γ , ratio of specific heats = 1.4.
- R = rho, density.

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PI90	SAC42	SAC43	SAC44
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1.300	1.200	1.100	1.000
1.400	1.200	1.000	1.200
1.500	1.200	1.000	1.000
1.600	1.200	1.200	1.000
1.800	1.200	1.000	1.200
2.000	1.200	1.000	1.000
2.200	1.200	1.000	1.200
2.400	1.200	1.000	.645
2.600	.630	.761	.781
1.200	.630	.897	.810
1.300	.630	1.000	.845
1.400	.712	1.000	.841
1.500	.734	1.000	.820
1.600	.760	1.000	.904
1.800	.734	1.000	.782
2.000	.758	1.000	.783
2.200	.784	.648	.533
2.400	.814	.791	.602
2.600	.836	.847	.630
1.200	.836	1.000	.716
1.300	.836	1.000	.741
1.400	.832	1.000	.723
1.500	.832	1.000	.700
1.600	.816	1.000	.665
1.200	.816	.632	.509
1.300	.874	.750	.546
1.400	.824	.845	.581
1.500	.824	.820	.609
1.600	.850	.882	.620
1.800	.850	1.000	.645
2.000	.834	1.000	.672
2.200	.854	1.000	.637
2.400	.868	1.000	.637
2.600	.852	1.000	.644
1.200	.874	.613	.484
1.300	.826	.729	.516
1.400	.870	.821	.545
1.500	.862	.895	.544
1.600	.828	.954	.583
1.800	.874	1.000	.618
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PISTON AREA INCHES SET PUMP. VARIABLE DISCHARGE PRESSURE

PISTON	1.000	1.175	1.350	1.525	1.700	1.875	2.050	2.225	2.400	2.575	2.750	2.925	3.100	3.275	3.450	3.625	3.800	3.975	4.150	4.325	4.500	4.675	4.850	5.025	5.200	5.375	5.550	5.725	5.900	6.075	6.250	6.425	6.600	6.775	6.950	7.125	7.300	7.475	7.650	7.825	8.000	8.175	8.350	8.525	8.700	8.875	9.050	9.225	9.400	9.575	9.750	9.925	10.100	10.275	10.450	10.625	10.800	10.975	11.150	11.325	11.500	11.675	11.850	12.025	12.200	12.375	12.550	12.725	12.900	13.075	13.250	13.425	13.600	13.775	13.950	14.125	14.300	14.475	14.650	14.825	15.000	15.175	15.350	15.525	15.700	15.875	16.050	16.225	16.400	16.575	16.750	16.925	17.100	17.275	17.450	17.625	17.800	17.975	18.150	18.325	18.500	18.675	18.850	19.025	19.200	19.375	19.550	19.725	19.900	20.075	20.250	20.425	20.600	20.775	20.950	21.125	21.300	21.475	21.650	21.825	22.000	22.175	22.350	22.525	22.700	22.875	23.050	23.225	23.400	23.575	23.750	23.925	24.100	24.275	24.450	24.625	24.800	24.975	25.150	25.325	25.500	25.675	25.850	26.025	26.200	26.375	26.550	26.725	26.900	27.075	27.250	27.425	27.600	27.775	27.950	28.125	28.300	28.475	28.650	28.825	29.000	29.175	29.350	29.525	29.700	29.875	30.050	30.225	30.400	30.575	30.750	30.925	31.100	31.275	31.450	31.625	31.800	31.975	32.150	32.325	32.500	32.675	32.850	33.025	33.200	33.375	33.550	33.725	33.900	34.075	34.250	34.425	34.600	34.775	34.950	35.125	35.300	35.475	35.650	35.825	36.000	36.175	36.350	36.525	36.700	36.875	37.050	37.225	37.400	37.575	37.750	37.925	38.100	38.275	38.450	38.625	38.800	38.975	39.150	39.325	39.500	39.675	39.850	40.025	40.200	40.375	40.550	40.725	40.900	41.075	41.250	41.425	41.600	41.775	41.950	42.125	42.300	42.475	42.650	42.825	43.000	43.175	43.350	43.525	43.700	43.875	44.050	44.225	44.400	44.575	44.750	44.925	45.100	45.275	45.450	45.625	45.800	45.975	46.150	46.325	46.500	46.675	46.850	47.025	47.200	47.375	47.550	47.725	47.900	48.075	48.250	48.425	48.600	48.775	48.950	49.125	49.300	49.475	49.650	49.825	50.000	50.175	50.350	50.525	50.700	50.875	51.050	51.225	51.400	51.575	51.750	51.925	52.100	52.275	52.450	52.625	52.800	52.975	53.150	53.325	53.500	53.675	53.850	54.025	54.200	54.375	54.550	54.725	54.900	55.075	55.250	55.425	55.600	55.775	55.950	56.125	56.300	56.475	56.650	56.825	57.000	57.175	57.350	57.525	57.700	57.875	58.050	58.225	58.400	58.575	58.750	58.925	59.100	59.275	59.450	59.625	59.800	59.975	60.150	60.325	60.500	60.675	60.850	61.025	61.200	61.375	61.550	61.725	61.900	62.075	62.250	62.425	62.600	62.775	62.950	63.125	63.300	63.475	63.650	63.825	64.000	64.175	64.350	64.525	64.700	64.875	65.050	65.225	65.400	65.575	65.750	65.925	66.100	66.275	66.450	66.625	66.800	66.975	67.150	67.325	67.500	67.675	67.850	68.025	68.200	68.375	68.550	68.725	68.900	69.075	69.250	69.425	69.600	69.775	69.950	70.125	70.300	70.475	70.650	70.825	71.000	71.175	71.350	71.525	71.700	71.875	72.050	72.225	72.400	72.575	72.750	72.925	73.100	73.275	73.450	73.625	73.800	73.975	74.150	74.325	74.500	74.675	74.850	75.025	75.200	75.375	75.550	75.725	75.900	76.075	76.250	76.425	76.600	76.775	76.950	77.125	77.300	77.475	77.650	77.825	78.000	78.175	78.350	78.525	78.700	78.875	79.050	79.225	79.400	79.575	79.750	79.925	80.100	80.275	80.450	80.625	80.800	80.975	81.150	81.325	81.500	81.675	81.850	82.025	82.200	82.375	82.550	82.725	82.900	83.075	83.250	83.425	83.600	83.775	83.950	84.125	84.300	84.475	84.650	84.825	85.000	85.175	85.350	85.525	85.700	85.875	86.050	86.225	86.400	86.575	86.750	86.925	87.100	87.275	87.450	87.625	87.800	87.975	88.150	88.325	88.500	88.675	88.850	89.025	89.200	89.375	89.550	89.725	89.900	90.075	90.250	90.425	90.600	90.775	90.950	91.125	91.300	91.475	91.650	91.825	92.000	92.175	92.350	92.525	92.700	92.875	93.050	93.225	93.400	93.575	93.750	93.925	94.100	94.275	94.450	94.625	94.800	94.975	95.150	95.325	95.500	95.675	95.850	96.025	96.200	96.375	96.550	96.725	96.900	97.075	97.250	97.425	97.600	97.775	97.950	98.125	98.300	98.475	98.650	98.825	99.000	99.175	99.350	99.525	99.700	99.875	100.050	100.225	100.400	100.575	100.750	100.925	101.100	101.275	101.450	101.625	101.800	101.975	102.150	102.325	102.500	102.675	102.850	103.025	103.200	103.375	103.550	103.725	103.900	104.075	104.250	104.425	104.600	104.775	104.950	105.125	105.300	105.475	105.650	105.825	106.000	106.175	106.350	106.525	106.700	106.875	107.050	107.225	107.400	107.575	107.750	107.925	108.100	108.275	108.450	108.625	108.800	108.975	109.150	109.325	109.500	109.675	109.850	110.025	110.200	110.375	110.550	110.725	110.900	111.075	111.250	111.425	111.600	111.775	111.950	112.125	112.300	112.475	112.650	112.825	113.000	113.175	113.350	113.525	113.700	113.875	114.050	114.225	114.400	114.575	114.750	114.925	115.100	115.275	115.450	115.625	115.800	115.975	116.150	116.325	116.500	116.675	116.850	117.025	117.200	117.375	117.550	117.725	117.900	118.075	118.250	118.425	118.600	118.775	118.950	119.125	119.300	119.475	119.650	119.825	120.000	120.175	120.350	120.525	120.700	120.875	121.050	121.225	121.400	121.575	121.750	121.925	122.100	122.275	122.450	122.625	122.800	122.975	123.150	123.325	123.500	123.675	123.850	124.025	124.200	124.375	124.550	124.725	124.900	125.075	125.250	125.425	125.600	125.775	125.950	126.125	126.300	126.475	126.650	126.825	127.000	127.175	127.350	127.525	127.700	127.875	128.050	128.225	128.400	128.575	128.750	128.925	129.100	129.275	129.450	129.625	129.800	129.975	130.150	130.325	130.500	130.675	130.850	131.025	131.200	131.375	131.550	131.725	131.900	132.075	132.250	132.425	132.600	132.775	132.950	133.125	133.300	133.475	133.650	133.825	134.000	134.175	134.350	134.525	134.700	134.875	135.050	135.225	135.400	135.575	135.750	135.925	136.100	136.275	136.450	136.625	136.800	136.975	137.150	137.325	137.500	137.675	137.850	138.025	138.200	138.375	138.550	138.725	138.900	139.075	139.250	139.425	139.600	139.775	139.950	140.125	140.300	140.475	140.650	140.825	141.000	141.175	141.350	141.525	141.700	141.875	142.050	142.225	142.400	142.575	142.750	142.925	143.100	143.275	143.450	143.625	143.800	143.975	144.150	144.325	144.500	144.675	144.850	145.025	145.200	145.375	145.550	145.725	145.900	146.075	146.250	146.425	146.600	146.775	146.950	147.125	147.300	147.475	147.650	147.825	148.000	148.175	148.350	148.525	148.700	148.875	149.050	149.225	149.400	149.575	149.750	149.925	150.100	150.275	150.450	150.625	150.800	150.975	151.150	151.325	151.500	151.675	151.850	152.025	152.200	152.375	152.550	152.725	152.900	153.075	153.250	153.425	153.600	153.775	153.950	154.125	154.300	154.475	154.650	154.825	155.000	155.175	155.350	155.525	155.700	155.875	156.050	156.225	156.400	156.575	156.750	156.925	157.100	157.275	157.450	157.625	157.800	157.975	158.150	158.325	158.500	158.675	158.850	159.025	159.200	159.375	159.550	159.725	159.900	160.075	160.250	160.425	160.600	160.775	160.950	161.125	161.300	161.475	161.650	161.825	162.000	162.175	162.350	162.525	162.700	162.875	163.050	163.225	163.400	163.575	163.750	163.925	164.100	164.275	164.450	164.625	164.800	164.975	165.150	165.325	165.500	165.675	165.850	166.025	166.200	166.375	166.550	166.725	166.900	167.075	167.250	167.425	167.600	167.775	167.950	168.125	168.300	168.475	168.650	168.825	169.000	169.175	169.350	169.525	169.700	169.875	170.050	170.225	170.400	170.575	170.750	170.925	171.100	171.275	171.450	171.625	171.800	171.975	172.150	172.325	172.500	172.675	172.850	173.025	173.200	173.375	173.550	173.725	173.900	174.075	174.250	174.425	174.600	174.775	174.950	175.125	175.300	175.475	175.650	175.825	176.000	176.175	176.350	176.525	176.700	176.875	177.050	177.225	177.400	177.575	177.750	177.925	178.100	178.275	178.450	178.625	178.800	178.975	179.150	179.325	179.500	179.675	179.850	180.025	180.200	180.375	180.550	180.725	180.900	181.075	181.250	181.425	181.600	181.775	181.950	182.125	182.300	182.475	182.650	182.825	183.000	183.175	183.350	183.525	183.700	183.875	184.050	184.225	184.400	184.575	184.750	184.925	185.100	185.275	185.450	185.625	185.800	185.975	186.150	186.325	186.500	186.675	186.850	187.025	187.200	187.375	187.550	187.725	187.900	188.075	188.250	188.425	188.600	188.775	188.950	189
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13. ABSTRACT The jet pump is a device that compresses low-pressure fluid by means of a high energy jet. This device is simple, reliable and light weight, and is attractive for various uses including eventual application to boundary layer control for aircraft. In this study the theoretical flow of a perfect gas through constant pressure and constant area jet pumps is predicted by analyzing the equations of continuity, energy and momentum. Of particular interest is the effect of heating the high energy jet. The parameters describing optimum heated jet pumps are determined. The ideal pumps presented herein represent upper performance limits for actual devices. Also included is a complete description and design of a facility for testing heated jet pumps. Specific test configurations are analyzed, and performance curves are illustrated. These curves facilitate comparison between theory and experiment.			

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14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Jet						
Pump						
Heated						
Boundary Layer						
Flow						
Mixing						
Compressible						
Subsonic						
Tube						
Design						
Control						
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14. KEY WORDS: Key words are technically meaningful terms or short phrases that characterize a report and may be used as index entries for cataloging the report. Key words must be selected so that no security classification is required. Identifiers, such as equipment model designation, trade name, military project code name, geographic location, may be used as key words but will be followed by an indication of technical context. The assignment of links, roles, and weights is optional.

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